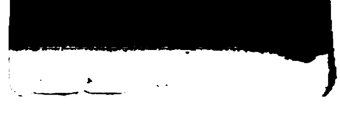

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OF

STEAM ENGINEERING

BY

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INTRODUCTION.

In the production of this work the great aim of the author has been to produce a book that would embrace the entire field of the science of steam engineering, and to present that important science, in all of its various branches, in its simplest possible form, so as to bring it within the understanding of engineers of ordinary education, and thus give them an equal chance in competition with those who are more fortunately situated with regard to the possession of knowledge in the higher branches of learning. To accomplish this desirable object, the rules and formulæ appertaining to the mathematics and geometry of steam engineering have been reduced to their utmost simplicity, and brought within the easy comprehension of every engineer who understands the simple rules of ordinary arithmetic.

It has also been the purpose and aim of the author to embrace, in this same volume, all the information necessary to enable engineers to pass a most successful and rigid examination by inspection officers; to enable engineers to fill positions of inspectors; to become experts in making evaporative and calorimeter boiler tests, and indicating steam engines, adjusting valves and valve-gear, and draughting plans and specifications for the construction of steam boilers; in short, to become proficient in all the arts and sciences of steam engineering—stationary, locomotive and marine.

The measure of success that has attended the efforts of the author toward the accomplishment of these objects, must be determined by those who follow the profession of steam engineering. To them, then, this volume is respectfully dedicated.

THE AUTHOR.

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A LIBRARY OF STEAM ENGINEERING.

CHAPTER I.

MATHEMATICS OF STEAM ENGINEERING.

The profession of steam engineering is composed of three separate and distinct branches, namely: "The Theory of Steam Engineering, The Practice of Steam Engineering, and The Mathematics of Steam Engineering." The latter is the fundamental principle upon which the whole science of steam engineering is based; and the engineer who is deficient in this indispensable branch of his profession can neither be a first-class, reliable, safe nor an economical engineer. Therefore, to the completion of his education, as an engineer, it is absolutely indispensable that he should be familiar with, at least, the ordinary rules of arithmetic, if not with the higher branches of mathematics.

There is another reason why every man who intends to follow the profession of steam engineering should thoroughly familiarize himself with the mathematics of steam engineering. The U. S. Government has set the example, and the various states in the union are rapidly following in the passage of laws providing for the examination and licensing of steam engineers. The time therefore is fast approaching when no man who is deficient in this important science will be permitted to take charge of any steam plant, or follow the profession of steam engineering anywhere in the United States. To this may be added still another, and by no means unimportant reason. The engineer who wishes to attain a high position in his profession and command the highest standard of wages can never realize his expectations without a thorough knowledge of the mathematics of steam engineering. But the great difficulty which invariably confronts the student in his initial effort to acquire that knowledge is familiar to every engineer. To obviate that difficulty it is essential that a course of instruction for the student should be arranged in its utmost simplicity and kept throughout within the comprehension of every engineer of the most ordinary education. To accomplish that desirable object is the purpose and aim of the course of instruction laid down in this work; and as this work is devoted largely to the mathematics of steam engineering, it is essential that the student should familiarize himself with at least the ordinary terms employed in mathematics; as these terms are employed universally in all works

on steam engineering as well as in all others devoted to mathematical science. We will therefore begin with addition, the first branch of mathematics.

ADDITION.

The Sum.—When two or more numbers are added together, the total is called the sum. Thus:

```
1890
1891
1892
----
5673 The Sum.
```

SUBTRACTION.

The Remainder.—When a smaller number is taken from a greater number that which remains is called the remainder. Thus:

```
250 Minuend.
175 Subtrahend.
---
 75 Remainder.
```

MULTIPLICATION.

Multiplicand, Multiplier, Product.—The multiplicand is the number to be multiplied; the multiplier is the number with which to multiply the multiplicand, the result is called the product. Thus:

```
 248 Multiplicand.
  25 Multiplier.
----
1240
496
----
6200 The Product.
```

LONG DIVISION.

Divisor, Dividend, Quotient, Remainder.—The divisor is the number by which the dividend is divided; the dividend is the number divided by the divisor; the quotient is the number that represents the number of times the divisor is contained in the dividend; the remainder is that part of the dividend remaining which has not been divided by the divisor. Thus:

```
             Dividend.
Divisor. 25 ) 7859 ( 314 Quotient.
              75
              --
               35
               25
               --
               109
               100
               ---
                 9 Remainder.
```

DECIMAL FRACTIONS.

In addition to the above, decimal fractions, square root and cube root enter largely into the mathematics of steam engineering and particularly decimal fractions, which branch of mathematics is constantly called into play and the necessity of which is met by the student at the very threshold of steam engineering. Therefore, before the student of steam engineering enters upon the study of the science of the profession he *must*, if he desires to make any progress whatever, familiarize himself with the branch of mathematics called decimal fractions.

CONSTANTS

3.1416.

Such numbers as 3.1416 and .7854 are called constants and as these particular constants enter more largely into steam engineering calculations than any other constants they will be here explained.

TO DETERMINE THE CIRCUMFERENCE OF A CIRCLE.

The constant 3.1416 is universally employed in determining the circumference of things circular in form, and the operation is performed by multiplying the constant 3.1416 by the diameter of the circle in inches and the product will give the circumference of the circle in inches.

Example.—Taking a circle 2 inches in diameter and we have:

3.1416	Constant.
2	Diameter of circle.
6.2832	Circumference of circle in inches and decimals of an inch.

TO DETERMINE THE DIAMETER OF A CIRCLE.

RULE.—Divide the circumference of the circle in inches by 3.1416 *and the quotient will give the diameter in inches.*

Example. — Taking a circle 6.2832 inches in circumference and we have:

3.1416) 6.2832 (2 Diameter of circle in inches.
6.2832

.7854

As to the constant .7854 a more extended explanation will be required. There is hardly an engineer who has not frequently made use of that constant in making his calculations for determining the area of things circular in form; yet very few of those who constantly use it have any knowledge as to what it represents, or from what or whence it is derived. In the first place, the student is informed that it repre-

sents the area of any circle whose diameter is unity. That is, whose diameter is, for example, one inch, one foot, one yard, one mile or one anything in diameter. But as it is almost invariably employed in determining the number of square inches in a given circle, the rules here laid down will be arranged so as to be made applicable to that purpose.

When we square the diameter of any circle we obtain for our answer the number of circular inches in the circle; and as each circular inch contains .7854 of a square inch, by multiplying the number of circular inches contained in a circle of any given diameter by .7854 we obtain the number of square inches contained in the circle.

The area of a circular inch as expressed in decimal fractions is .7854 of a square inch, which means that the area is one inch by .7854 of an inch in dimensions and not .7854 of an inch square as many engineers erroneously consider it. A square inch has four equal sides, each one inch in length, and all fractions of a square inch when expressed in decimals, contain a length of one inch, and the decimals, whatever they may be, express the width.

TO REDUCE THE AREA OF A CIRCLE TO A PERFECT SQUARE.

To reduce decimals of a square inch or any number of square inches to a perfect square it is necessary to extract the square root of the decimals, or of the number of square inches.

A 4-inch circle contains 12.5664 square inches. As thus expressed, its dimensions are one inch in height and 12.5664 inches in length; and if reduced to a perfect square would contain four equal sides each 3.5449+ inches in length.

Performing the operation we have:

```
         12.5664 ( 3.5449+ inches, length of side of square.
          9
         ---
      65) 356
          325
         ----
     704) 3164
          2816
         -----
    7084) 34800
          28336
         ------
   70889) 646400
          638001
```

DECIMAL PARTS OF A SQUARE INCH.

The following figures contain an illustration of the dimensions of parts of square inches as expressed in decimal fractions.

Fig. 1 contains .7854 of a square inch as expressed in decimal fractions.

Fig. 2 contains .5 of a square inch as expressed in decimal fractions.
Fig. 3 contains .25 of a square inch as expressed in decimal fractions.

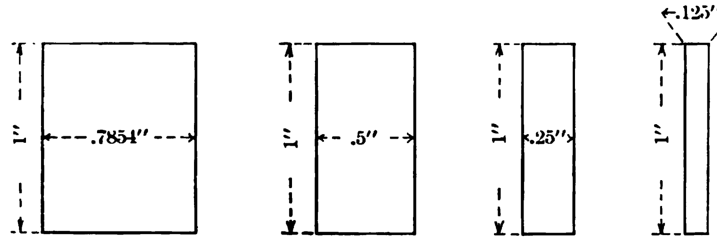


Fig. 1 Fig. 2 Fig. 3 Fig. 4

Fig. 4 contains .125 of a square inch as expressed in decimal fractions.

HOW THE CONSTANT .7854 IS OBTAINED.

The next step is to show how the constant .7854 is obtained. To illustrate one of the methods, a circle is drawn one inch in diameter, as shown in Fig. 5, which is divided into 24 equal parts, and the parts then arranged in the shape of a parallelogram, as shown in Fig. 6. A circle, one inch in diameter, as shown in Fig. 5, has a circumference of 3.1416 inches, and if arranged as shown in Fig. 6 will have a width of ½ inch, and a length of 1.5708 inches nearly, according to the number of pieces into which the circle is divided; the greater the number of pieces the nearer the length will approach one-half of the circumference of the circle, and the nearer the width will approach one-half of its diameter. One-half of the circle is represented by the wavy line at the top of Fig. 6, and the other half is represented by the wavy line at the bottom of the figure. Although this illustration is often given as being correct, it will be observed that it is not absolutely correct, and

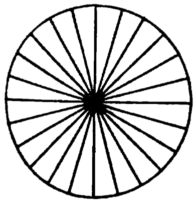


Fig. 5

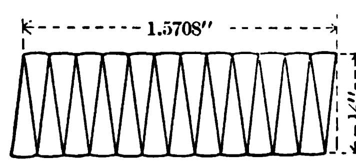


Fig. 6

can not be made so, no matter into how many pieces the circle is divided; it can, however, be made near enough correct to answer the purpose of illustration, hence the following rule.

RULE.—Multiply the length of the parallelogram by its width and the product will give the area.

Example.—Let 1.5708 inches equal length of Fig. 6, .50 (fifty one-hundredths) of an inch equal width of Fig. 6. Then we have:

$$\begin{array}{r} 1.5708 \\ .50 \\ \hline .785400 \end{array}$$ **Area of a circle one inch in diameter.**

Now as it is well known that all ciphers to the right of the last decimal neither increase nor diminish the value of the decimals, the ciphers in the answer are dropped for convenience, and we have for our purpose the much-used, and little-known, mysterious constant .7854.

It is well for the student to know also that the constant .7854 itself is not strictly correct, but it is near enough for all practical purposes, so much so that its adoption by mathematicians has become universal.

Another method of showing how the constant .7854 is obtained is by geometrical rule according to the following illustration.

A sector of a circle, as shown in Fig. 7, is the part included by the two radii and the intercepted arc A B C.

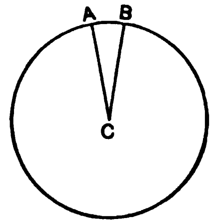


Fig. 7

RULE.—Ascertain the area of a sector by multiplying the distance circumferentially between the two points in the arc, A B, as shown in Fig. 7 by the radius C, and dividing the product by 2.

As the circle is one inch in diameter its circumference is 3.1416 inches; and if the circle be divided according to the previous illustration (Fig. 5) into 24 sectors, the distance of each, circumferentially, will be 3.1416÷24=.1309 inch; which multiplied by the radius C A or C B, $\frac{1}{2}$ inch, or reduced to a decimal fraction .5 (five-tenths) of an inch, would be .1309×.5=.06545, which divided by 2 would be .06545÷2= .032725, which is the area of a sector. As the circle, in this case, is divided into 24 sectors, we multiply the area of one sector by 24 and the product will give the area of the circle, and consequently the constant .7854. Thus:

$$\begin{array}{r} .032725 \\ 24 \\ \hline 0\,130900 \\ 0\,65450 \\ \hline 0.785400 \end{array}$$

As in the former case, the ciphers being dropped for convenience, we have the constant .7854.

TO FIND THE AREA OF A CIRCLE.

Among the first things the student is required to learn is the rule for determining the area of a circle or a valve, which is as follows:

RULE.—Square the diameter in inches or decimals of an inch of the circle or valve, and multiply the product by .7854.

What is meant by the term squaring the diameter of a circle or a valve is this: It is multiplying the diameter by the diameter. For example: If a circle or valve is 3.25 inches in diameter, we say 3.25 times 3.25. Thus:

```
   3.25
   3.25
   ----
   1625
   650
  9 75
-------
10.5625  The square of the diameter
         of the valve.
```

MULTIPLICATION OF DECIMAL FRACTIONS.

In multiplying decimal fractions always point off as many decimals in the product as there are decimals in the multiplicand and multiplier. In the above case there are two in each; we therefore point off four decimals, counting, always, from right to left, and all figures to the left of the decimal point are whole numbers, and all figures to the right of the decimal point are the fractional part of unity or of one whole number.

The next step is to multiply the square of the diameter of the circle or valve by the constant .7854. Thus:

```
   10.5625
     .7854
   -------
    422500
   528125
  845000
 7 39375
----------
8.29578750  Area of circle or valve
            in square inches.
```

THE CORRECT DIAMETER OF A VALVE.

In taking the diameter of a valve, engineers are cautioned not to be governed by the valve opening leading into the valve chamber—a mistake commonly made, but they must be governed by the *smallest* diameter of the valve's bearing on the seat. A valve may cover an

opening, say three inches in diameter, and the valve may have a bearing all around the opening a quarter of an inch from the opening, in which case the diameter of the valve would be 3½ inches instead of 3 inches.

Example.—Taking a valve whose smallest diameter of its bearing on the seat is 4 inches.

Then we have: $4 \times 4 \times .7854 = 12.5664$, area of valve. Thus: 4 times 4 are 16, and 16 times .7854 are 12 and $\frac{5664}{10,000}$ square inches. Or, we may put it in the following shape. First, square the diameter of the valve thus:

$$\begin{array}{r} 4 \\ 4 \\ \hline 16 \end{array}$$

Then multiply the decimal .7854 by 16—the square of the diameter of the valve. Thus:

$$\begin{array}{rl} .7854 & \\ 16 & \\ \hline 4\,7124 & \\ 7\,854 & \\ \hline 12.5664 & \text{Area of the valve in square inches.} \end{array}$$

TO DETERMINE THE REQUIRED DIAMETER OF A VALVE OF ANY GIVEN AREA.

In determining the required diameter of a valve, the first thing to be done is to ascertain the amount of area required to make the valve effective for the purpose for which it is intended. After the amount of area required is ascertained, proceed as follows:

RULE.—Divide the number of square inches in the area by .7854, and then extract the square root of the quotient.

Example.—Taking a valve required to have an area of 12.5664 square inches, what must be its diameter? Then we have:

$$\begin{array}{rrl} .7854) & 12.5664\,(16 & \text{Square of the diameter of the valve.} \\ & 7\,854 & \\ \hline & 4\,7124 & \\ & 4\,7124 & \\ \hline \end{array}$$

The next step is to extract the square root of the quotient, which in this case represents the square of the diameter of the valve. Thus:

$$\begin{array}{rl} \dot{1}6\,(4 & \text{Diameter of a valve required to contain an area of 12.5664 square inches.} \\ 16 & \\ \hline \end{array}$$

Performing the operation in a shorter method we have:

$$\sqrt{\frac{12.5664}{.7854}}=4 \text{ Diameter of the valve in inches.}$$

TO DETERMINE THE CIRCUMFERENCE OF A VALVE.

RULE.—Multiply the smallest diameter of its bearing on the seat by the constant 3.1416, *the product will give the circumference of the valve or of a circle.*

Example.—Taking a valve having a diameter of 3.5 inches, and we have:

$$\begin{array}{r} 3.1416 \\ 3.5 \\ \hline 1\,57080 \\ 9\,4248 \\ \hline 10.99560 \end{array} \text{ Circumference of the valve in inches.}$$

TO FIND THE DIAMETER REQUIRED FOR A VALVE OF ANY GIVEN CIRCUMFERENCE.

RULE.—Divide the given circumference by the constant 3.1416, *the quotient will give the required diameter of a valve or circle.*

Example.—Taking a valve or circle having a circumference of 10.99560 inches, then we have:

$$\begin{array}{rl} 3.14160)\ 10.99560\ (3.5 & \text{This quotient represents diameter required in inches.} \\ 9\,42480 & \\ \hline 1\,570800 & \\ 1\,570800 & \\ \hline \end{array}$$

ANNEXATION OF CIPHERS IN DIVISION.

It will be noticed that a cipher has been added to the decimal in the divisor. This is done under the rule to avoid confusion between the whole numbers and the decimal numbers in the quotient. In dividing with decimal fractions, or in other words, in using decimal fractions when performing the operation of division, the divisor and dividend should each contain the same number of decimals. If one contains more than the other, a sufficient number of ciphers should be annexed to the lesser whichever it may be, either the divisor or the dividend, to equal the number of decimals in the other, then, so long as there is any figure or cipher in the dividend which has not been brought down in performing the operation, or in other words, so long as the entire dividend has not been exhausted the quotient produced by dividing the remainder will be a whole number. But as soon as

the dividend has become exhausted and a cipher has been borrowed to continue the operation, a decimal point must be put immediately after the last whole number in the quotient, and all the figures which follow the decimal point to the right are decimals. Thus:

```
20) 343 (17
    20
    ---
    143
    140
    ---
      3 Remainder.
```

According to this we have 3 as a remainder and this represents a vulgar fraction $\frac{3}{20}$. Now if we wish to continue the operation until we have a sufficient number of decimals—four are generally sufficient for all practical purposes—or until there is no remainder, we proceed as follows:

```
20) 343 (17.15
    20
    --
    143
    140
    ---
     30  This cipher is borrowed and we therefore
     20      put a decimal point in the quotient,
     ---     and all figures which follow that point
    100      are decimals, and we continue to bor-
    100      row until there is no remainder.
    ---
```

TO REDUCE COMMON FRACTIONS TO DECIMALS.

In reducing a vulgar or common fraction to a decimal we divide the numerator by the denominator. The numerator is the number above the line and the denominator is the number below the line. Thus $\frac{3}{20}$ is called three-twentieths. In reducing it to a decimal we say: 20 into 3 goes no times, and we write a cipher in the quotient. Now as we are compelled to borrow a cipher we place a decimal point after the cipher and continue the operation until there is no remainder.

```
First operation:    20) 3 (0.

Second operation:   20) 3.0 (0.1
                        2 0
                        ---
                        1 0

Third operation:    20) 3.0 (0.15  Fifteen one hundredths.
                        2 0
                        ---
                        1 00
                        1 00
                        ----
```

Therefore $\frac{15}{100}$ is equal to $\frac{3}{20}$ and in decimal fractions is expressed thus: .15.

In mathematics of steam engineering all vulgar fractions should first be reduced to decimal fractions. For example: Taking a valve $3\frac{3}{4}$ inches in diameter, the first thing to be done is to reduce the $\frac{3}{4}$ to a decimal fraction. Thus:

```
4) 3.0 (0.75   This represents three-fourths of an
   2 8              inch; and the diameter of the
   ---              valve will be expressed, 3.75.
    20
    20
    --
```

We now square this diameter if we wish to determine the area. Thus:

```
      3.75
      3.75
     -----
      1875
     2 625
    11 25
   -------
   14.0625   Square of the diameter of the valve.
```

We now multiply the square of the diameter of the valve by the constant .7854 and the product will give the area of the valve in square inches. Thus:

```
      14.0625
        .7854
      -------
       562500
      703125
    1 125000
    9 84375
  -----------
  11.04468750   Area of valve.
```

It will be seen that we have four decimals in the multiplicand and four decimals in the multiplier, making eight decimals in both. We now count eight from right to left in the product and place a decimal point to the left of the eighth figure. Now all figures to the left of the decimal point are whole numbers and all figures to the right of the decimal point are fractions or decimals.

TO DETERMINE THE NUMBER OF CUBIC INCHES IN A CYLINDRICAL WEIGHT.

RULE.—Square the diameter and multiply .7854 by the product; then multiply the last product by the length of the weight.

Example.—Taking a cylindrical weight 8 inches in diameter and 12 inches in length, and we have:

.7854	
$8 \times 8 = 64$	Square of the diameter of the weight.
3 1416	
47 124	
50.2656	
12	Length of the weight.
100 5312	
502 656	
603.1872	Number of cubic inches contained in the weight.

TO DETERMINE THE NUMBER OF POUNDS CONTAINED IN A CAST-IRON WEIGHT.

RULE.—Multiply the number of cubic inches contained in the weight by .2607—the weight of a cubic inch of cast-iron.

Example.—Taking a cast-iron weight 8 inches in diameter and 12 inches in length and we have:

.7854	
$8 \times 8 = 64$	Square of diameter.
3 1416	
47 124	
50.2656	
12	Length of weight.
100 5312	
502 656	
603.1872	Number of cubic inches in the weight.
.2607	Weight of a cubic inch of cast-iron.
42223104	
36 191232	
120 63744	
157.25090304	Number of pounds contained in the weight.

TO DETERMINE THE NUMBER OF POUNDS CONTAINED IN A WROUGHT-IRON LEVER OF A SAFETY-VALVE.

RULE.—Multiply the number of cubic inches of iron, in the lever by .2817; *which is the weight of one cubic inch of wrought-iron.*

Example.—Taking a lever of uniform section, for example, having a length of 40 inches, a width of 2.5 inches and a thickness of .5 of an inch, we then multiply the length by the width, the product by the thickness, and the last product by .2817. Thus:

```
   40    Length of lever.
   2.5   Width of lever.
  ----
  200
  80
 -----
 100.0
    .5   Thickness of lever.
 -----
 50.00   Cubic inches.
```

Hence there are 50 cubic inches of iron contained in the lever. The next operation is to find the weight of the lever. Thus:

```
.2817    Weight in decimals of a pound of one cubic inch of wrought-iron.
   50    Number of cubic inches contained in lever.
-------
14.0850  Number of pounds weight contained in the lever, which it will
         be observed are 14 and 8 one hundredths of a pound.
```

TO DETERMINE THE NUMBER OF POUNDS WEIGHT CONTAINED IN A WROUGHT-IRON VALVE SPINDLE.

RULE.—First square the diameter of the spindle; then multiply the product by .7854, then multiply the product of that operation by the length of the spindle; then multiply the last product by .2817; the answer will give the number of pounds contained in the spindle.

Example.—Taking a spindle $\frac{7}{8}$ of an inch in diameter and 8 inches in length, and we have, First:

```
8) 7.0 (0.875   The diameter of the spindle in decimals
   64            of an inch.
   --
    60
    56
    --
     40
     40
     --
```

Then:

```
           .875
           .875
          -----
          4375
         6125
        7000
        -------
        .765625   Square of the diameter of the spindle.
          .7854
        -------
        3062500
       3828125
      6125000
     5359375
     -----------
```

Am't carried forward, .6013218750

Am't brought forward, .6013218750
8

4.8105750000 Number of cubic inches of iron in the spindle. As the ciphers have no value they are dropped, and that leaves but 6 decimals.
.2817

33674025
4810575
38484600
9621150

1.3551389775 Number of pounds contained in the spindle. It will be observed that it weighs a fraction over one pound and one-third of a pound.

TO DETERMINE THE WEIGHT OF THE VALVE WITHOUT WEIGHING IT.

RULE.—Immerse the valve in a vessel containing enough water to submerge the valve, and just large enough in diameter to contain the valve, so as to show as much displacement of water as possible; note the rise of water in the vessel, and from the measurement of this rise determine the number of cubic inches of water displaced by the submerging of the valve, and that will represent the number of cubic inches of metal in the valve.

If the valve is of cast-iron, multiply the number of cubic inches of water displaced by .2607; but if the valve is of brass, multiply the number of cubic inches of water displaced by .3194.

Example.—Taking a vessel 6″ in diameter, and 8″ in height, and containing 6″ of water before the valve is submerged therein, and 7½″ of water after the valve is submerged, we then have a rise of 1½″ of water in a vessel 6″ in diameter. Then 6 times 6 are 36, the square of the diameter of the space occupied by the water. Then:

.7854
6×6= 36 Square of the diameter of the vessel.

47124
23562

28.2744 Number of square inches in the diameter.
1.5 Rise of water in the vessel.

14 13720
28 2744

42.41160 Number of cubic inches contained in the rise of water by submerging the valve.

Now suppose the valve to be of brass, then we have:

42.4116
.3194

1696464
3817044
424116
12 72348

13.54626504 Number of pounds contained in the valve.

In order that the student may not be misled, it will be here observed that this is a large valve, say 6″ in diameter, with wings for guides extending down into the valve opening below the seat.

TO DETERMINE THE WEIGHT OF A CAST-IRON BALL OR SPHERE.

RULE.—Multiply the cube of the diameter of the ball by .5236, *and then multiply the last product by* .2607, *the answer will give the weight in pounds.*

Example.—Let 8″ equal diameter of a spherical cast-iron weight.

Then we have $8 \times 8 \times 8 \times .5236 \times .2607 = 69.88+$ lbs.

Or $8^3 \times .5236 \times .2607 = 69.88+$ lbs.

Putting it in the ordinary way, we have:

8	Diameter of sphere.
8	Diameter of sphere.
64	Square of diameter of sphere.
8	Diameter of sphere.
512	Cube of diameter.
.5236	
1 0472	
5 236	
261 80	
268.0832	Area of sphere in square inches.
.2607	Weight of a cubic inch of cast-iron.
18765824	
16 084992	
53 61664	
69.88929024 lbs.	Weight of the sphere.

The student is now prepared to advance one step and consider the adjustment of the safety-valve mathematically, hence, the next chapter will be devoted to that subject.

CHAPTER II.

THE SAFETY-VALVE MATHEMATICALLY CONSIDERED.

The first step necessary in the elucidation of this subject is to adjust, by the aid of a diagram, a safety-valve as it is found in practice, and then to explain in detail the various points connected therewith. Recourse will be had to skeleton drawings because of their simplicity and because they come more readily within the comprehension of those in need of the information here imparted. The following drawing shows a practical illustration of the adjustment of the safety-valve and it will constitute the foundation upon which the instructions in this chapter are based.

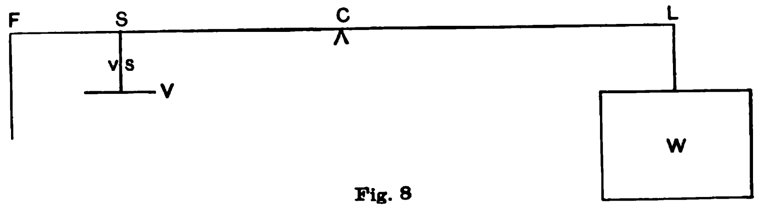


Fig. 8

NAMES OF DIFFERENT PARTS OF THE LEVER.

F represents the fulcrum—the point on which the lever turns.

A common error committed by many engineers is to call the distance from F to S the fulcrum when it is nothing of the kind.

S represents the length of the short arm of the lever; and the short arm of the lever is that portion of the lever between F and S.

C represents the center of gravity of the lever, that is the point on which the lever will balance on a knife-edge. In a lever of uniform section the center of gravity will be at or near the middle in the length of the lever. If the lever is larger at the fulcrum and tapers toward the outer end, the center of gravity will be nearer the fulcrum.

L represents the long arm of the lever, and therefore the long arm of the lever is that portion of the lever between F and L.

Vs represents the valve spindle.

V represents the valve.

CENTER OF GRAVITY OF THE LEVER.

As the center of gravity of the lever plays an important part in the adjustment of all lever safety-valves, mathematically, and without

a consideration of which no lever valve can be accurately adjusted mathematically, it is therefore necessary that the student should thoroughly understand what use is to be made of the center of gravity of the lever in making his calculations in determining the pressure required to raise the valve, the distance the weight is to be placed from the fulcrum to allow the valve to rise at any given pressure, to determine the number of pounds a weight must contain to be placed a given distance on the lever from the fulcrum; in short, in making all calculations connected with the adjustment of the valve, mathematically, or in determining the weight or dimensions of any and all portions of the safety valve. All this may seem complicated to the beginner, and yet the whole matter connected with the center of gravity of the lever can be summed up in the following words: The only use to be made of the center of gravity of the lever is to determine the amount of pressure produced on the valve by the weight of the lever. It is well known that the pressure produced on the valve by the lever depends upon the weight of the lever, the distance of the center of gravity of the lever from the fulcrum, and the length of the short arm of the lever. Hence, to determine the total pressure—not per square inch—the lever produces on the valve, we multiply the distance in inches of the center of gravity of the lever from the fulcrum by the weight of the lever in pounds, and then divide the product by the length of the short arm of the lever in inches, the quotient will give the total pressure on the valve produced by the weight of the lever. To illustrate we will make use of the diagram (Fig. 9).

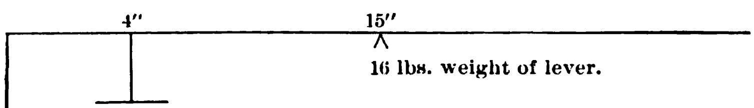


Fig. 9

4'' represents the length of the short arm of the lever in inches or the distance from the fulcrum to the center of the valve.

15'' represents the distance in inches of the center of gravity of the lever from the fulcrum.

The weight of the lever is 16 pounds, as shown in Fig. 9, and to determine the amount of pressure it produces on the valve we will proceed according to the rule above laid down. Then we have:

15'' Distance of center of gravity from the fulcrum.
Weight of the lever. 16 lbs.
90
15
Length of short arm of the lever. 4'') 240
60 lbs. The total pressure produced on the valve by the weight of the lever.

This is the same as though a weight of 60 pounds was placed on the top of the valve. If it is required to determine the amount of pressure per square inch produced on the valve by the weight of the lever, we divide the total pressure produced on the valve by the area of the valve, that is by the number of square inches in the valve, and the quotient will give the pressure per square inch produced on the valve by the weight of the lever.

To illustrate this subject a little further, the student is informed that he must treat the center of gravity of the lever the same as he would a weight equal to the weight of the lever, hung on the lever at the center of gravity. In the case above mentioned the distance of the center of gravity from the fulcrum is 15 inches and the weight of the lever is 16 pounds, and the total pressure produced on the valve by the weight of the lever is 60 pounds. Now, in addition to that, if a weight of 16 pounds is placed on the lever—that is the center of the weight—15 inches from the fulcrum that weight would produce a pressure on the valve of 60 pounds also, and the two combined would produce a pressure of twice 60 pounds or 120 pounds.

We are now prepared to advance another step, and for the purpose of simple illustration we will continue to employ the diagram method, and adjust a safety-valve complete as found in every day practice.

TO DETERMINE THE PRESSURE PER SQUARE INCH REQUIRED TO RAISE THE FOLLOWING VALVE.

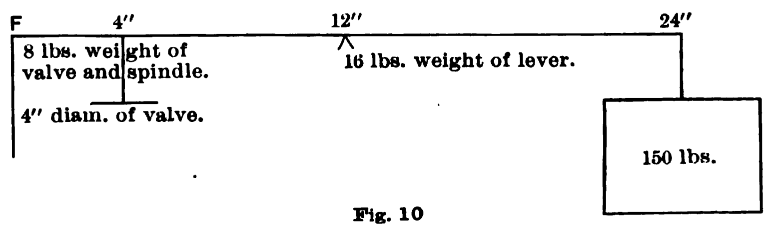


Fig. 10

The information required to determine the pressure per square inch necessary to raise a safety-valve as shown in Fig. 10, is shown in the diagram, with the exception of the area of the valve, which should be determined in order to make a diagram complete. In order then to aid the student and to simplify this subject we will construct a diagram for each operation required to be made, and place only such figures in the diagram as may be required to perform each particular operation. The first thing, then, to make the diagram complete is to determine the area of the valve, for which purpose we will employ the following diagram.

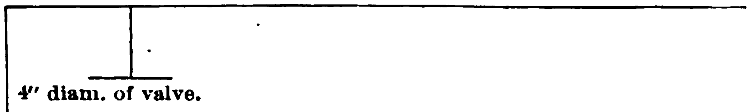


Fig. 11

TO DETERMINE THE AREA OF THE VALVE.

RULE.—Square the diameter of the valve, that is, multiply the diameter by the diameter; then multiply the product by .7854, the last product will give the area of the valve.

Example.—Taking a valve 4'' in diameter, we have:

$$4 \times 4 \times .7854 = 12.5664 \text{ square inches. Area of the valve.}$$

To put it in simple form, we have:

```
      4
      4
   ----
     16
  .7854
 ------
  47124
  7854
-------
12.5664  Area of the valve in square inches.
```

This enables us to make the diagram complete as follows:

TO DETERMINE THE PRESSURE PER SQUARE INCH REQUIRED TO RAISE THE FOLLOWING VALVE.

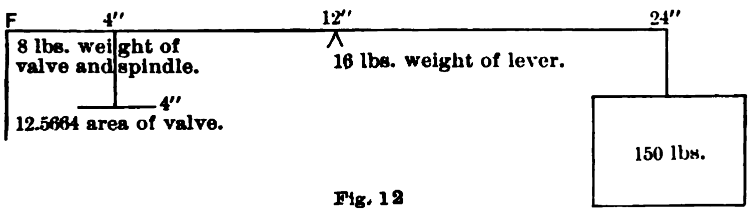


Fig. 12

F represents the fulcrum.

4'' represents the length of the short arm of the lever, or, the distance of the center of the valve from the fulcrum.

12'' represents the distance of the center of gravity of the lever from the fulcrum.

24'' represents the length of the long arm of the lever, and also the distance of the weight from the fulcrum.

8 lbs. represents the weight of the valve and spindle.

16 lbs. represents the weight of the lever.

4'' represents the diameter of the valve.

12.5664 represents the area of the valve in square inches.

150 lbs. represents the number of pounds contained in the weight.

The first thing the student is required to do is to familiarize himself thoroughly with either of the following rules in determining the pressure, per square inch, required to raise the valve.

RULES FOR DETERMINING THE PRESSURE REQUIRED TO RAISE THE VALVE.

RULE I.—First, divide the distance of the center of gravity of the lever from the fulcrum by the length of the short arm of the lever, and then multiply the quotient by the weight of the lever. Call this product "Product No. 1."

Second, divide the distance in inches the center of the weight is hung on the lever from the fulcrum by the distance in inches the center of the valve is from the fulcrum, and then multiply the quotient by the number of pounds contained in the weight, and call this product "Product No. 2."

Third, add the weight of the valve and the spindle and Products Nos. 1 and 2 together and call this " The Sum."

Fourth, divide "The Sum" by the area of the valve, and the quotient will give the pressure, per square inch, required to raise the valve.

Where there are fractions in the weights or distances the following method will be found the simplest:

RULE II.—First, multiply the weight of the valve and spindle in pounds by the distance in inches of the center of the valve from the fulcrum, and call this product " Product No. 1."

Second, multiply the distance in inches of the center of gravity of the lever from the fulcrum by the weight of the lever in pounds, and call this product " Product No. 2."

Third, multiply the distance in inches of the center of the weight from the fulcrum by its weight in pounds, and call this product "Product No. 3."

Fourth, add Products Nos. 1, 2 and 3 together and call the answer " The Sum."

Fifth, multiply the area of the valve by the distance in inches of its center from the fulcrum, and call this product " Product No. 4."

Sixth, divide " The Sum" by "Product No. 4" and the quotient will give the pressure per square inch required to raise the valve.

These rules, it will be observed, differ very materially from each other, yet the answer in both cases will be exactly the same.

We will now proceed according to the first rule to determine the pressure required to raise the valve as adjusted in Fig. 12, by the aid of the following diagrams.

FIRST OPERATION.

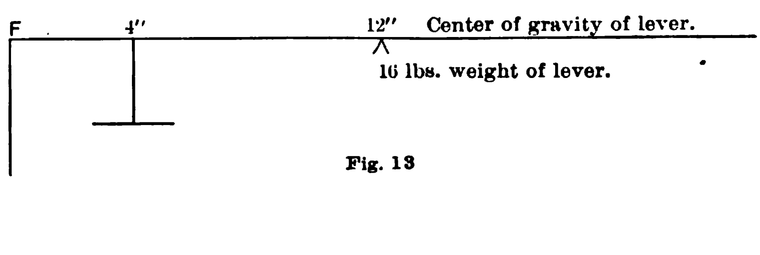


Fig. 13

Figure 13, it will be observed, contains only such figures as are required to perform the first operation according to the first paragraph of the first rule, as follows:

"*First, divide the distance of the center of gravity of the lever from the fulcrum by the length of the short arm of the lever, and then multiply the quotient by the weight of the lever, and call the product 'Product No. 1.'*"

Example.—Let 4 inches equal length of short arm of the lever.

Let 12 inches equal distance of center of gravity of the lever from the fulcrum.

Let 16 pounds equal weight of the lever.

Then we have:

$$\frac{12}{4}\times16=48 \text{ lbs.}$$ Pressure produced on the valve by the weight of the lever.

Putting it in different form, we have:

$$\begin{array}{r|l} 4 & 12 \\ \hline & 3 \\ & 16 \\ \hline & 48 \text{ lbs.} \end{array}$$

This we set aside and call it "Product No. 1."

We will perform the second operation by the aid of the following diagram, in determining the pressure produced on the valve by the weight

SECOND OPERATION.

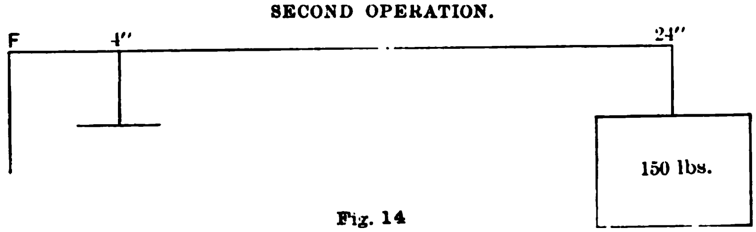


Fig. 14

Figure 14 contains only such figures as are required to perform the operation as laid down in the second paragraph of the first rule, which is as follows:

"*Second, divide the distance in inches of the center of the weight hung on the lever from the fulcrum by the length of the short arm of the lever in inches, and then multiply the quotient by the number of pounds contained in the weight, and call this product 'Product No. 2.'*"

Example.—Let 4 inches equal length of short arm of the lever.

Let 24 inches equal distance of the center of the weight from the fulcrum.

Let 150 pounds equal the number of pounds contained in the weight.

Then we have:

$$\frac{24}{4}\times 150 = 900 \text{ lbs.}$$ **Pressure produced on the valve by the weight of the weight.**

Putting it in different form, we have:

$$\begin{array}{r} 4)\,24 \\ \hline 6 \\ 150 \\ \hline 900 \end{array}$$

Which we set aside and call it "Product No 2" 900 lbs.

We will perform the next operation by determining the pressure produced on the valve seat by the weight of the valve and spindle.

THIRD OPERATION.

Fig. 15

It will be observed that there is but one figure connected with the diagram proper, and that represents the weight of the valve and spindle, and as that weight is directly over the valve seat, no operation is required to determine the pressure produced on the seat, but simply follow the directions as laid down in the third paragraph of the first rule, which is as follows:

"*Third, add the weight of the valve and spindle to Products Nos.* 1 *and* 2 *and call this 'The Sum.'*"

Example.—Let 8 pounds equal the weight of the valve and spindle.
Let 48 pounds equal "Product No. 1."
Let 900 pounds equal "Product No. 2."
And we have:

$$\begin{array}{r} 8 \\ 48 \\ 900 \\ \hline 956 \end{array}$$

Total pressure on the valve seat produced by the weight of valve, spindle, lever and weight, which we set aside and call it "The Sum" 956 lbs.

FOURTH OPERATION.

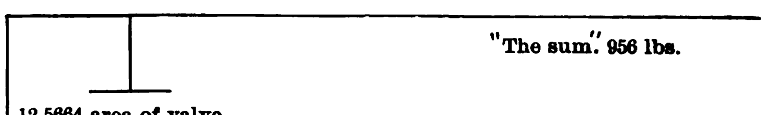


Fig. 16

Here, also, all the figures required to perform the operation are in

the diagram; so we will proceed according to directions laid down in the fourth paragraph of the first rule as follows:

"*Fourth divide 'The Sum' by the area of the valve and the quotient will give the pressure, per square inch, required to raise the valve.*"

Example.—Let 12.5664 square inches equal the area of the valve.
Let 956 pounds equal "The Sum."

Then we have:

```
12.5664) 956.0000 (76.07+   Pounds per square inch required
         879 648                  to raise the valve.
         -------
          76 3520
          75 3984
          -------
            953600
            879648
            ------
```

Putting the example in shorter form, we have:

$$\frac{956.0000}{12.5664} = 76.07+ \text{ lbs.}$$

We will now proceed to perform the various operations in accordance with directions contained in Rule 2.

TO DETERMINE THE PRESSURE, PER SQUARE INCH, REQUIRED TO RAISE THE FOLLOWING VALVE, ACCORDING TO RULE 2.

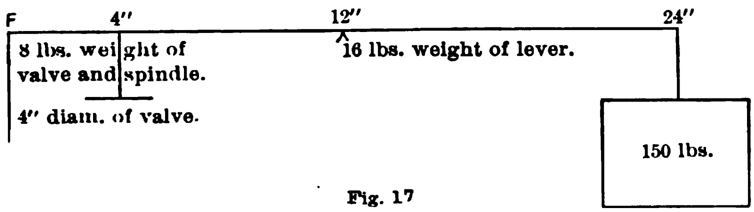


Fig. 17

It will be observed that the area of the valve is not given in the diagram. The first thing then to be done to complete the diagram is to determine the area of the valve, and the method for determining that is the same in Rule 2 as it is in Rule 1, so we will begin with the use of the following diagram:

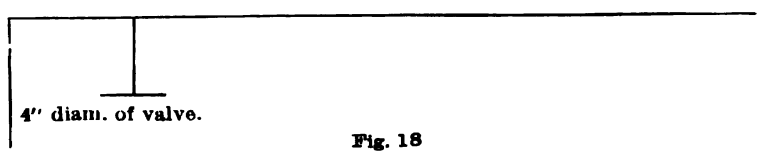


Fig. 18

TO DETERMINE THE AREA OF THE VALVE.

RULE.—Square the diameter of the valve, that is, multiply the diameter by the diameter, then multiply the product by .7854, the last product will give the area of the valve.

Example.—Taking a valve 4 inches in diameter, as shown in Fig. 18, we have:

$4 \times 4 \times .7854 = 12.5664$ square inches—area of the valve.

Putting it in simple form, we have:

$$\begin{array}{r} 4 \\ 4 \\ \hline 16 \\ .7854 \\ \hline 4\,7124 \\ 7\,854 \\ \hline 12.5664 \end{array}$$

Area of the valve in square inches.

This enables us to complete the diagram as follows:

TO DETERMINE THE PRESSURE PER SQUARE INCH REQUIRED TO RAISE THE FOLLOWING VALVE.

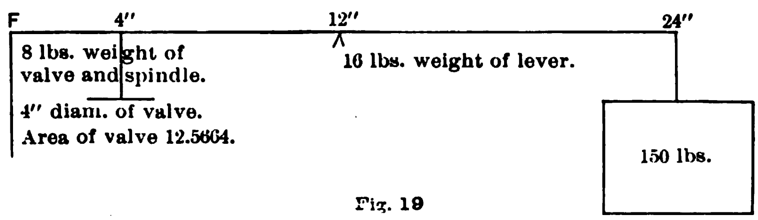


Fig. 19

F represents the fulcrum.

4″ represents the length of the short arm of the lever, or the distance in inches of the center of the valve from the fulcrum.

12″ represents the distance in inches of the center of gravity of the lever from the fulcrum.

24″ represents the length in inches of the long arm of the lever, and also the distance of the weight from the fulcrum.

8 lbs. represents the weight of the valve and spindle.

16 lbs. represents the weight of the lever.

150 lbs. represents the weight of the weight.

4″ represents the diameter of the valve in inches.

12.5664 represents the area of the valve in square inches.

We will now proceed according to Rule 2 to determine the pressure per square inch required to raise the valve as adjusted in Fig. 19.

FIRST OPERATION.

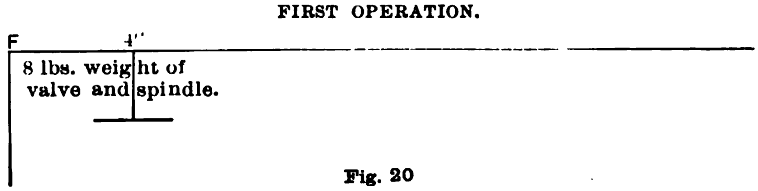


Fig. 20

RULE.—Multiply the weight of the valve and spindle by the length of the short arm of the lever, and call this product "Product No. 1."

Example.—Let 4 inches equal length of the short arm of the lever.
Let 8 pounds equal weight of valve and spindle.

Then we have $8\times4=32$, "Product No. 1."

Putting it in simpler form, we have:

$$\begin{array}{r} 8 \\ \underline{4} \\ 32 \end{array} \quad \text{"Product No. 1."}$$

SECOND OPERATION.

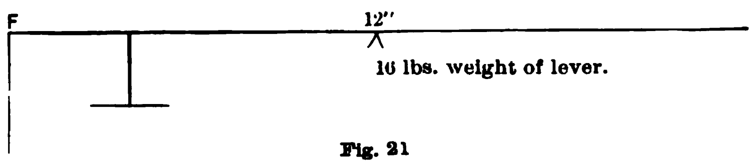


Fig. 21

RULE.—Multiply the distance in inches of the center of gravity of the lever from the fulcrum by the weight of the lever in pounds, and call the product "Product No. 2."

Example.—Let 12 inches equal distance of center of gravity of the lever from the fulcrum.
Let 16 pounds equal weight of lever.

Then we have $12\times16=192$, "Product No. 2."

Putting it in different form, we have:

$$\begin{array}{r} 12 \\ \underline{16} \\ 32 \\ \underline{16} \\ 192 \end{array} \quad \text{"Product No. 2."}$$

THIRD OPERATION.

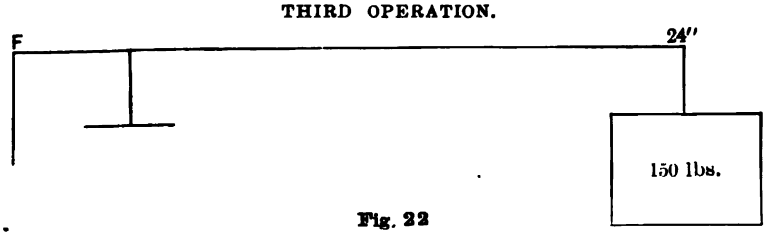


Fig. 22

RULE.—Multiply the number of pounds contained in the weight by the distance in inches of its center from the fulcrum, and call this product "Product No. 3."

Example.—Let 24 inches equal distance of center of weight from the fulcrum.

Let 150 pounds equal weight of the weight.

Then we have 150×24=3600, "Product No. 3."

Putting it in the following form, we have:

$$\begin{array}{r} 150 \\ 24 \\ \hline 600 \\ 300 \\ \hline 3600 \end{array} \quad \text{"Product No. 3."}$$

FOURTH OPERATION.

As this operation has reference to the result of the operations relating to the preceding diagrams, it needs no diagram for the purpose of illustration. So we will proceed according to the following rule:

RULE.—Add Products Nos. 1, 2 *and* 3 *together and call the answer "The Sum."*

Example.—Let 32 equal "Product No. 1."

Let 192 equal "Product No. 2."

Let 3600 equal "Product No. 3."

Then we have 32+192+3600=3824. "The Sum."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 32 \\ 192 \\ 3600 \\ \hline 3824 \end{array} \quad \text{"The Sum."}$$

FIFTH OPERATION.

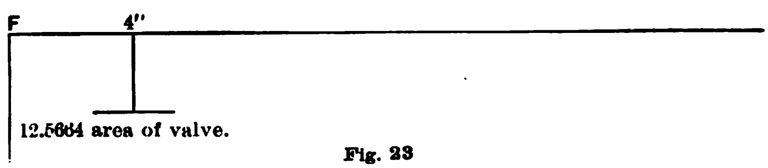


Fig. 23

RULE.—Multiply the area of the valve in square inches by the distance in inches of its center from the fulcrum, and call this product "Product No. 4."

Example.—Let 4 inches equal distance of center of valve from the fulcrum.

Let 12.5664 square inches equal area of the valve.

Then we have 12.5664×4=50.2656. "Product No. 4."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 12.5664 \\ 4 \\ \hline 50.2656 \end{array} \text{ "Product No. 4."}$$

SIXTH OPERATION.

As the various answers requiring the use of diagrams have been obtained by the aid of such diagrams, this operation does not require any diagram. So we will proceed at once according to the rule.

RULE.—Divide "The Sum" by "Product No. 4," and the quotient will give the pressure, per square inch, required to raise the valve as adjusted in Fig. 19.

Example.—Let 3824 equal "The Sum."
Let 50.2656 equal "Product No. 4," then we have:

$$\frac{3824}{50.2656} = 76.07 + \text{ Pounds pressure per square inch required to raise the valve.}$$

Performing the operation in the ordinary manner, we have:

$$\begin{array}{l} 50.2656)\ 3824.0000\ (76.07+ \text{ Pounds pressure per square inch required to raise the valve.} \\ 3518\,592 \\ \hline 305\,4080 \\ 301\,5936 \\ \hline 3\,814400 \\ 3\,518592 \\ \hline \end{array}$$

TO DETERMINE THE DISTANCE ON THE LEVER TO PLACE THE GIVEN WEIGHT.

Having determined the pressure required to raise the valve, we will now proceed to determine the distance a given weight will have to be placed on the lever to allow the valve to rise at a given pressure; and as these problems will gradually become more and more complicated, as we proceed toward the end of the chapter, we will employ a valve requiring the use of small numbers, and in this way keep the work down to its utmost simplicity. We will, therefore, construct a small valve, and first determine the pressure per square inch, according to Rule 2, required to raise it, and all calculations made hereafter to the end of this chapter, will have reference to this valve for the purpose of showing the correctness of the rule as well as the correctness of the calculations made.

TO DETERMINE THE PRESSURE PER SQUARE INCH REQUIRED TO RAISE THE FOLLOWING VALVE.

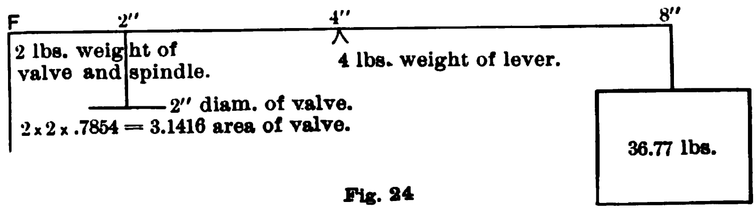


Fig. 24

Proceeding according to Rule 2, we have:

First, the distance in inches of the center of the valve from the fulcrum multiplied by the weight of the valve and spindle in pounds, 2×2=" Product No. 1." **4**

Second, the distance in inches of the center of gravity of the lever from the fulcrum multiplied by the weight of the lever in pounds, 4×4=" Product No. 2." **16**

Third, the distance in inches of the center of the weight from the fulcrum multiplied by the number of pounds contained in the weight, 8×36.77 equal "Product No 3." **294.16**

Fourth, Products 1, 2 and 3 added together="The Sum," **314.16**

Fifth, the area of the valve, multiplied by the distance in inches of its center from the fulcrum equals "Product No. 4" $2\times2\times.7854=3.1416$ area of valve, and $3.1416 \times2=6.2832$, " Product No. 4."

Sixth, "The Sum" divided by "Product No. 4"

$$\frac{\text{"The Sum"}\ 314.16}{\text{"Product No. 4"}\ 6.2832}=50 \text{ pounds.}$$

This, it will be observed, gives 50 pounds pressure per square inch required to raise the valve.

Putting it in another form, we have:

$$\frac{(2\times2)+(4\times4)+(8\times36.77)}{2\times2\times.7854\times2}=50 \text{ pounds.}$$

Performing the operation, we have:

2″	Distance in inches of center of valve from fulcrum.
2	Weight of valve and spindle in pounds.
4	" Product No. 1."

4″	Distance of center of gravity of lever from fulcrum.
4	Weight of lever in pounds.
16	" Product No. 2."

36.77	Weight of weight in pounds.
8″	Distance of weight from fulcrum.
294.16	"Product No. 3."

Adding Products Nos. 1, 2 and 3 together, we have:

4	
16	
294.16	
314.16	"The Sum."

Performing the operation below the line in the example, we have:

2″	Diameter of valve.
2″	Diameter of valve.
4	Square of the diameter of valve.
.7854	A constant.
3.1416	Area of valve.
2″	Distance of center of valve from fulcrum.
6.2832	"Product No. 4."

Dividing "The Sum" by "Product No. 4," we have:

6.2832) 314.1600 (50 Pounds pressure per square inch required to raise the valve.
314 160
0

TO DETERMINE THE DISTANCE IN INCHES A GIVEN WEIGHT IS TO BE PLACED ON THE LEVER FROM THE FULCRUM TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE PER SQUARE INCH.

To determine the distance from the fulcrum the weight in Fig. 25 will be required to be placed on the lever to allow the valve to rise at a pressure of 50 pounds per square inch.

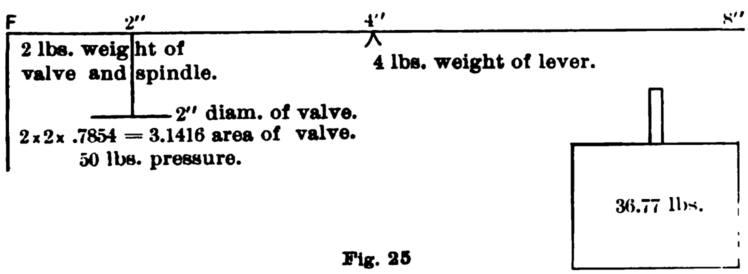


Fig. 25

RULE.—First, multiply the distance in inches the center of valve is from the fulcrum by the weight of the valve and spindle in pounds, and call the product "Product No. 1."

Second, multiply the distance in inches the center of gravity of the lever is from the fulcrum by the weight of the lever in pounds, and call the product "Product No. 2."

Third, add Products Nos. 1 and 2 together and call the answer "The Sum."

Fourth, multiply the distance in inches the center of the valve is from the fulcrum by the area of the valve in square inches and multiply the product by the required pressure per square inch, and call the last product "Product No. 3."

Fifth, subtract "The Sum" from "Product No. 3" and call the answer "The Remainder."

Sixth, divide "The Remainder" by the number of pounds contained in the given weight and the quotient will give the distance in inches the given weight is to be placed on the lever from the fulcrum to allow the valve to rise at the given pressure.

F represents the fulcrum.

2″ represents the distance in inches of the center of valve from the fulcrum.

4″ represents the distance in inches of center of gravity of the lever from the fulcrum.

2 lbs. represents the weight of the valve and spindle.

4 lbs. represents the weight of the lever.

2″ represents the diameter of the valve.

3.1416 represents the area of the valve in square inches.

50 lbs. represents the given pressure per square inch.

36.77 lbs. represents the weight of the weight.

FIRST OPERATION.

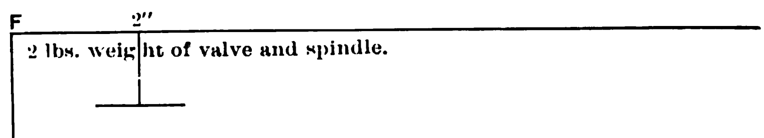


Fig. 26.

RULE.—Multiply the distance the center of the valve is from the fulcrum by the weight of the valve and spindle and call this product "Product No. 1."

Example.—Let 2″ equal distance of center of valve from fulcrum.
Let 2 lbs. equal weight of valve and spindle.

Then we have: $2 \times 2 = 4''$ "Product No. 1."

In the ordinary form, we have:

$$\begin{array}{r} 2 \\ 2 \\ \hline 4 \end{array}$$ "Product No. 1."

SECOND OPERATION.

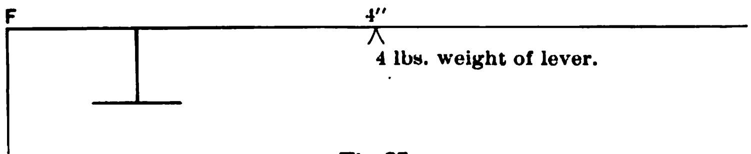


Fig. 27

RULE.—Multiply the distance the center of gravity of the lever is from the fulcrum by the weight of the lever and call this product "Product No. 2."

Example.—Let 4″ equal distance of center of gravity of the lever from the fulcrum.

Let 4 lbs. equal weight of the lever.

Then we have: $4 \times 4 = 16$, "Product No. 2."

In the ordinary form, we have:

$$\begin{array}{r} 4 \\ 4 \\ \hline 16 \end{array} \text{ "Product No. 2."}$$

THIRD OPERATION.

As this operation embodies only the operations performed in the two preceding examples, a diagram is not necessary for illustration.

RULE.—Add Products Nos. 1 and 2 together and call the answer "The Sum."

Example.—Let 4 equal "Product No. 1."

Let 16 equal "Product No. 2."

Then we have: $4 + 16 = 20$, "The Sum."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 16 \\ 4 \\ \hline 20 \end{array} \text{ "The Sum."}$$

FOURTH OPERATION.

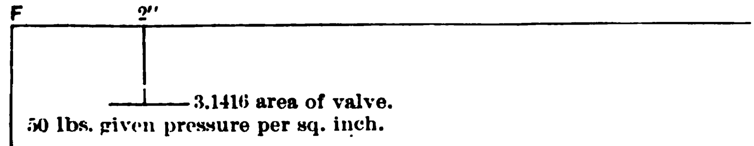


Fig. 28

RULE.—Multiply the area of the valve by the distance of center of the valve from the fulcrum, then multiply the product by the given pressure per square inch, required to raise the valve, and call the last product "Product No. 3."

Example.—Let 3.1416 square inches equal area of the valve.
Let 2″ equal distance of center of valve from fulcrum.
Let 50 lbs. equal given pressure per square inch required to raise the valve.

Then we have: $3.1416 \times 2 \times 50 = 314.16$, "Product No. 3."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 3.1416 \\ 2 \\ \hline 6.2832 \\ 50 \\ \hline 314.1600 \end{array} \quad \text{"Product No. 3."}$$

FIFTH OPERATION.

NOTE.—This operation can not be illustrated by diagram.

RULE.—Subtract "The Sum" from "Product No. 3" and call the answer "The Remainder."

Example.—Let 314.16 equal "Product No. 3."
Let 20 equal "The Sum."

Then we have: $314.16 - 20 = 294.16$, "The Remainder."

Putting it in the ordinary form, we have:

$$\begin{array}{rl} 314.16 & \text{"Product No. 3."} \\ 20 & \text{"The Sum."} \\ \hline 294.16 & \text{"The Remainder."} \end{array}$$

NOTE.—The student will notice that "The Remainder" contains two decimals and that "The Sum" contains none—the number 20 being a whole number. In subtraction the units in the subtrahend must always be placed under the units in the minuend, the tens under tens, hundreds under hundreds, and so on. As there are no units in decimals, and as decimals begin with tenths which are followed toward the right by hundredths, thousandths and so on; the same care must be exercised in subtracting decimals as there is in the case of whole numbers, all the denominations in the subtrahend must be placed under their respective denominations in the minuend.

SIXTH OPERATION.

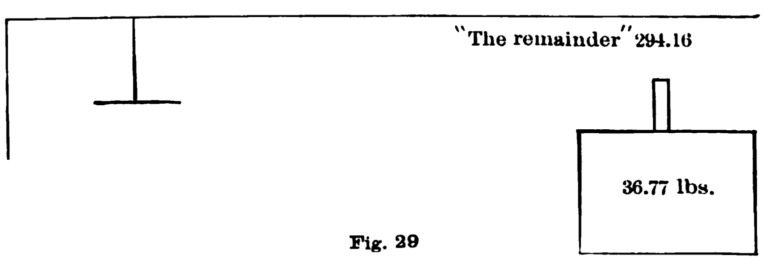


Fig. 29

RULE.—Divide "The Remainder" by the number of pounds contained in the weight, and the quotient will give the distance in inches the center of the weight will have to be placed on the lever to allow the valve to rise at the given pressure per square inch.

Example.—Let 294.16 equal " The Remainder."
Let 36.77 equal the number of pounds contained in the weight.

Then we have: $294.16 \div 36.77 = 8$ inches, the distance from the fulcrum the center of the weight will have to be placed to allow the valve to rise at a pressure of 50 pounds per square inch.

Putting it in the ordinary form, we have:

36.77) 294.16 (8 inches
294.16

To put the entire operation in condensed form:

Example.—Let 2 inches equal distance of center of valve from fulcrum.
Let 2 pounds equal weight of valve and spindle.
Let 4 inches equal distance of center of gravity of lever from the fulcrum.
Let 4 pounds equal weight of lever.
Let 3.1416 square inches equal area of valve.
Let 50 pounds equal pressure per square inch required to raise the valve.
Let 36.77 pounds equal the weight of the weight.

Then we have:

First, the distance of the center of the valve from the fulcrum multiplied by the weight of the valve and spindle, $2 \times 2 =$ " Product No. 1."	4
Second, the distance of the center of gravity of the lever from the fulcrum multiplied by the weight of the lever, $4 \times 4 =$ " Product No. 2."	16
Third, Products Nos. 1 and 2 added together = " The Sum."	20
Fourth, the distance of the center of the valve from the fulcrum multiplied by the area of the valve, and the product of this operation multiplied by the given pressure per square inch, $2 \times 3.1416 \times 50 =$ " Product No. 3."	314.16
Fifth, " The Sum " subtracted from " Product No. 3," $314.16 - 20 =$ " The Remainder."	294.16
Sixth, " The Remainder " divided by the number of pounds contained in the weight, $294.16 \div 36.77 = 8$ inches, the distance the weight must be placed from the fulcrum.	

Putting it in another form, we have:

$$\frac{(2 \times 3.1416 \times 50) - \overline{(2 \times 2) + (4 \times 4)}}{36.77} = 8 \text{ inches.}$$

Putting it in different form, we have:

$$3.1416 \times 2 \times 50 = 314.16$$
$$(2 \times 2) + (4 \times 4) = 20$$

Dividing by the weight, we have: 36.77) 294.16 (8 inches.
294.16

Performing the operation in the ordinary way, we have:

$2 \times 2 =$	4	"Product No. 1."
$4 \times 4 =$	16	"Product No. 2."
	20	"The Sum."
3.1416		Area of valve.
2″		Distance of fulcrum from center of valve.
6.2832		
50		Given pressure per square inch.
314.1600		"Product No. 3."
20		"The Sum."
294.16		"The Remainder."

We now divide "The Remainder" by the number of pounds contained in the weight, as follows:

36.77) 294.16 (8 inches. The distance the weight must be placed from the fulcrum to allow the valve to rise at a pressure of 50 pounds per square inch.
294.16

TO DETERMINE THE AMOUNT OF WEIGHT REQUIRED TO BE PLACED A GIVEN DISTANCE FROM THE FULCRUM TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE PER SQUARE INCH.

RULE.—First, multiply the distance the center of the valve is from the fulcrum by the weight of the valve and spindle, and call the product "Product No. 1."

Second, multiply the distance the center of gravity of the lever is from the fulcrum by the weight of the lever, and call the product "Product No. 2."

Third, add Products Nos. 1 and 2 together and call the answer "The Sum."

Fourth, multiply the distance the center of the valve is from the fulcrum by the area of the valve and multiply the product by the pressure, per square inch, required to raise the valve, and call the last product "Product No. 3."

Fifth, subtract "The Sum" from "Product No. 3" and call the answer "The Remainder."

Sixth, divide "The Remainder" by the distance, in inches, the weight is required to be placed from the fulcrum, and the quotient will give the number of pounds required in the weight.

To determine the number of pounds required in the weight to be placed on the lever, 8 inches from the fulcrum, in the following diagram, to allow the valve to rise at a pressure of 50 pounds per square inch.

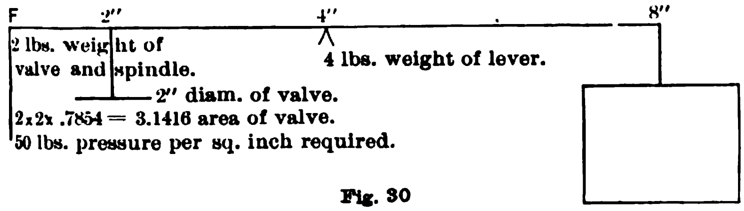


Fig. 30

F represents the fulcrum.

2″ represents the distance of the center of the valve from the fulcrum.

4″ represents the distance of the center of gravity of lever from the fulcrum.

8″ represents the distance the weight is to be placed from the fulcrum.

2 lbs. represents the weight of the valve and spindle.

4 lbs. represents the weight of the lever.

2″ represents the diameter of the valve.

3.1416 represents the area of the valve in square inches.

50 pounds represents the pressure per square inch required to be carried.

FIRST OPERATION.

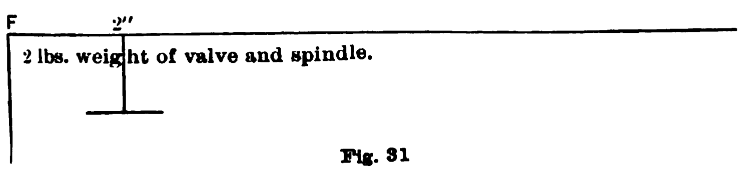


Fig. 31

RULE.—Multiply the distance the center of the valve is from the fulcrum by the weight of the valve and spindle and call the product "Product No. 1."

Example.—Let 2″ equal the distance of the center of the valve from the fulcrum.

Let 2 lbs. equal the weight of the valve and spindle.

Then we have: $2 \times 2 = 4$, "Product No. 1."

In the ordinary form, we have:

$$\begin{array}{r} 2 \\ 2 \\ \hline 4 \end{array}$$ "Product No. 1"

SECOND OPERATION.

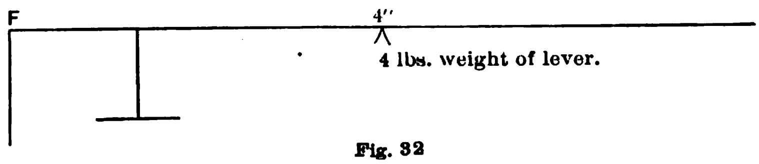


Fig. 32

RULE.—Multiply the distance the center of gravity of the lever is from the fulcrum, by the weight of the lever, and call the product "Product No. 2."

Example.—Let 4″ equal distance of center of gravity of lever from the fulcrum.
Let 4 lbs. equal weight of lever.

Then we have: $4 \times 4 = 16$, "Product No. 2."

THIRD OPERATION.

RULE.—Add Products Nos. 1 and 2 together and call the answer "The Sum."

Example.—Let 4 equal "Product No. 1."
Let 16 equal "Product No. 2."

Then we have: $4 + 16 = 20$, "The Sum."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 16 \\ 4 \\ \hline 20 \end{array} \text{ "The Sum."}$$

FOURTH OPERATION.

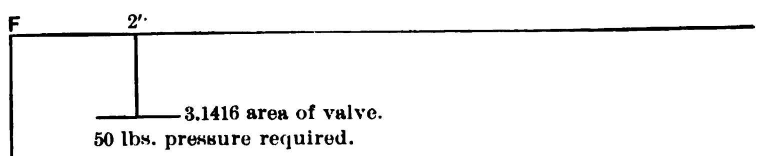


Fig. 33

RULE.—Multiply the distance the center of the valve is from the fulcrum by the area of the valve, and multiply the product by the pressure per square inch required to raise the valve, and call the last product, "Product No. 3."

Example.—Let 2″ equal distance of center of valve from the fulcrum.
Let 3.1416 square inches equal area of the valve.
Let 50 lbs. equal the given pressure.

Then we have: $2 \times 3.1416 \times 50 = 314.16$, "Product No. 3."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 3.1416 \\ 2 \\ \hline 6.2832 \\ 50 \\ \hline 314.1600 \end{array} \text{ "Product No. 3."}$$

FIFTH OPERATION.

RULE.—Subtract "The Sum" from "Product No. 3," and call the answer "The Remainder."

Example.—Let 314.16 equal "Product No. 3."
Let 20 equal "The Sum."

Then we have: 314.16—20=294.16, "The Remainder."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 314.16 \\ 20 \\ \hline 294.16 \end{array} \text{ "The Remainder."}$$

SIXTH OPERATION.

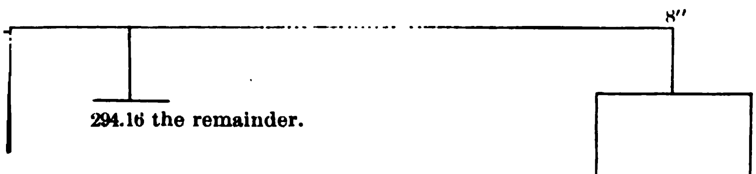


Fig. 34

RULE.—Divide "The Remainder" by the distance in inches the weight is required to be placed from the fulcrum and the quotient will give the number of pounds required in the weight to allow the valve to rise at a given pressure, which in this case is 50 pounds per square inch.

Example.—Let 294.16 equal "The Remainder."
Let 8″ equal the distance the weight is to be placed from the fulcrum.

Then we have: 294.16÷8=36.77 pounds.

Or, which is the same:

$$\frac{294.16}{8} = 36.77 \text{ pounds.}$$

Putting it in the ordinary form, we have:

```
8) 294.16 (36.77   Pounds required in the weight
   24                placed 8 inches from the ful-
   --                crum to allow the valve to rise
    54               at a pressure of 50 lbs. per
    48               square inch.
    --
     61
     56
     --
      56
      56
      --
```

NOTE.—The attention of the student is called to the last operation. It will be noticed that the dividend contains two decimals, while there is none in the divisor. In all cases of that kind whenever the first decimal is brought down and annexed to the remainder, a decimal point must be put after the last figure in the quotient, and all figures to the left of the decimal point are whole numbers, and all figures which follow to the right of the decimal point are decimals.

Putting the foregoing operations in condensed form:

Example.—Let 2 inches equal distance of center of valve from fulcrum.

Let 2 pounds equal weight of valve and spindle.

Let 4 inches equal distance of center of gravity of lever from the fulcrum.

Let 8 inches equal distance the weight is to be placed on the lever from the fulcrum.

Let 4 pounds equal weight of lever.

Let 3.1416 square inches equal area of the valve.

Let 50 pounds equal pressure per square inch required to raise the valve.

Then we have:

First, the distance of the center of the valve from the fulcrum multiplied by the weight of the valve and spindle, $2\times 2=$ "Product No. 1."	4
Second, the distance of the center of gravity of the lever from the fulcrum multiplied by the weight of the lever, $4\times 4=$ "Product No. 2."	16
Third, Products Nos. 1 and 2 added together equal "The Sum."	20
Fourth, the distance of the center of the valve from the fulcrum, multiplied by the area of the valve, and the product of this operation multiplied by the given pressure per square inch, $2\times 3.1416\times 50=$ "Product No. 3."	314.16
Fifth, "The Sum" subtracted from "Product No. 3," $314.16-20=$ "The Remainder."	294.16
Sixth, "The Remainder" divided by the distance in inches the weight is to be placed from the fulcrum, $29.416\div 8=$	

36.77 pounds required in the weight 8 inches from the fulcrum to allow the valve to rise at a pressure of 50 pounds per square inch.

Putting it in another form, we have:

$$\frac{(2\times 3.1416\times 50)-(2\times 2)+(4\times 4)}{8}=36.77 \text{ lbs.}$$

Putting it in different form, we have.

$$2\times 3.1416\times 50=314.16$$
$$(2\times 2)+(4\times 4)=\ 20$$

Dividing by the distance the weight 8) 294.16
is to be placed from the fulcrum. 36.77 lbs.

Performing the operation by the ordinary method, we have:

3.1416	Area of valve.
2″	Distance of center of valve from fulcrum.
6.2832	
50	Given pressure per square inch.
314.1600	"Product No. 3."
20	The sum to be subtracted.
294.16	"The Remainder."

We now divide the remainder by the distance in inches the weight is to be placed on the lever from the fulcrum.

Thus: 8) 294.16 (36.77 Pounds required in the weight.

```
8)294.16(36.77
  24
  --
   54
   48
   --
    61
    56
    --
     56
     56
     --
```

TO DETERMINE THE LENGTH OF THE SHORT ARM OF THE LEVER TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE PER SQUARE INCH.

RULE.—First, multiply the distance of the center of gravity of the lever from the fulcrum, by the weight of the lever, and call the product "Product No. 1."

Second, multiply the distance the weight is from the fulcrum by the number of pounds contained in the weight, and call this product "Product No. 2."

Third, add Products Nos. 1 and 2 together and call the answer "The Sum."

Fourth, multiply the area of the valve by the given pressure, per square inch, and deduct the weight of the valve and spindle from the product and call the answer "The Remainder."

Fifth, divide "The Sum" by "The Remainder," and the quotient will give the required length of the short arm of the lever in inches.

To determine the length required of the short arm of the lever in the diagram (Fig. 35), to allow the valve to rise at a pressure of 50 pounds per square inch.

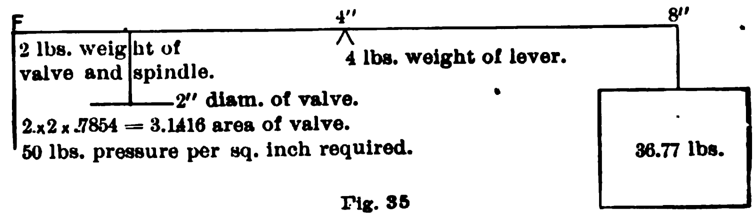


Fig. 35

F represents the fulcrum.

4" represents the distance of center of gravity of the lever from the fulcrum.

8" represents the distance the weight is from the fulcrum.

2 lbs. represents the weight of the valve and spindle.

4 lbs. represents the weight of the lever.

2" represents the diameter of the valve.

3.1416 represents the area of the valve in square inches.

50 lbs. represents the pressure per square inch required to raise the valve.

FIRST OPERATION.

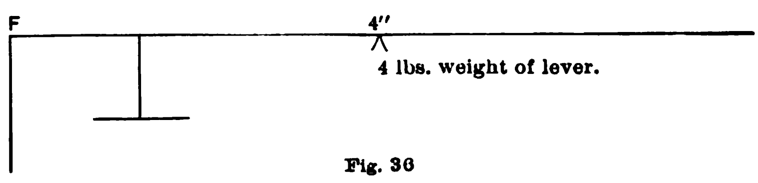


Fig. 36

RULE.—Multiply the distance of the center of gravity of the lever from the fulcrum by the weight of the lever and call the product "Product No. 1."

Example.—Let 4" equal distance of center of gravity of the lever from the fulcrum.

Let 4 lbs. equal weight of the lever.

Then we have: $4 \times 4 = 16$, "Product No. 1."

Putting it in the ordinary form:

4"	Distance of center of gravity of lever from fulcrum.
4	Weight of lever.
16	"Product No. 1."

SECOND OPERATION.

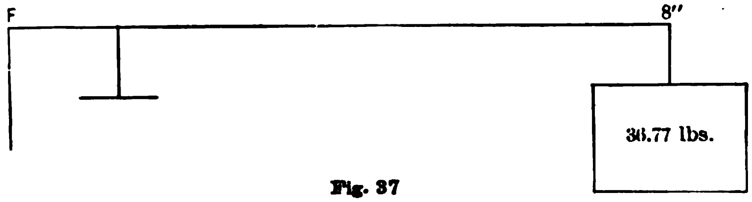


Fig. 37

RULE.—Multiply the number of pounds contained in the weight by the distance the weight is from the fulcrum and call this product " Product No. 2."

Example.—Let 8″ equal the distance the weight is from the fulcrum.

Let 36.77 lbs. equal the weight of the weight.

Then we have: $36.77 \times 8 = 294.16$, " Product No. 2."

Putting it in the ordinary form:

36.77	Weight of the weight.
8″	Distance of center of weight from fulcrum.
294.16	" Product No. 2."

THIRD OPERATION.

RULE.—Add Products Nos. 1 and 2 together and call the answer " The Sum."

Example.—Let 16 equal " Product No. 1."

Let 294.16 equal " Product No. 2."

Then we have: $16 + 294.16 = 310.16$, " The Sum."

Putting it in the ordinary form:

294.16	" Product No. 2."
16	" Product No. 1."
310.16	" The Sum."

FOURTH OPERATION.

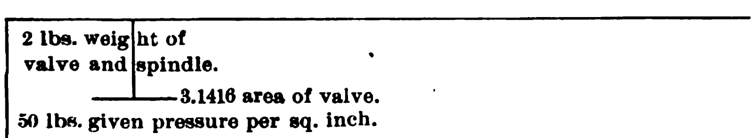


Fig. 38

RULE.—Multiply the area of the valve by the given pressure per square inch and then deduct the weight of the valve and spindle from the product and call the answer " The Remainder."

Example.—Let 50 lbs. equal the given pressure per square inch.
Let 2 lbs. equal the weight of the valve and spindle.
Let 3.1416 square inches equal the area of the valve.

Then we have $(3.1416 \times 50)-2=155.08$, "The Remainder."

Putting it in the ordinary form:

3.1416	Area of valve.
50	Given pressure.
157.0800	
2	Weight of valve and spindle.
155.08	"The Remainder."

FIFTH OPERATION.

RULE.—Divide "The Sum" by "The Remainder" and the quotient will give the length of the short arm of the lever, or in other words, the distance the center of the valve must be from the fulcrum to allow the valve to rise at a pressure of 50 *pounds per square inch.*

Example.—Let 155.08 equal "The Remainder."
Let 310.16 equal "The Sum."

Then we have: $310.16 \div 155.08=2$ inches. Required length of short arm of the lever.

Putting it in the ordinary form, we have:

155.08) 310.16 (2 inches. Required length of short arm of lever.
310.16

Putting the foregoing operation in condensed form:

Example.—Let 4 inches equal distance of center of gravity of lever from the fulcrum.
Let 4 pounds equal weight of lever.
Let 8 inches equal distance of weight from fulcrum.
Let 36.77 pounds equal weight of the weight
Let 3.1416 square inches equal area of valve.
Let 50 pounds equal the given pressure.
Let 2 pounds equal the weight of the valve and spindle.

Then we have:

$$\frac{(4\times4)+(36.77\times8)}{(3.1416\times50)-2}=2 \text{ inches.}$$ Length of short arm of lever.

Putting it in the ordinary form, we have:

$$4\times4=16$$
$$36.77\times8=294.16$$
$$310.16$$ "The Sum."

This completes the operation above the line in the preceding example and the answer is called "The Sum."

The next thing is to perform the operation below the line.

Thus

3.1416	Area of valve.
50	Given pressure.
157.0800	
2	Weight of valve and spindle to be subtracted.
155.08	"The Remainder."

This completes the operation below the line in the example. We now divide the answer obtained above the line by the answer obtained below the line. Thus:

"The Sum." "The Remainder."

$$310.16 \div 155.08 = 2 \text{ inches.}$$

Or:

155.08) 310.16 (2 inches. Length of short arm of lever.
310.16

TO DETERMINE THE DIAMETER OF THE VALVE REQUIRED TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE.

RULE.—First, multiply the distance of the center of the valve from the fulcrum by the weight of the spindle and estimated weight of the valve and call the product "Product No. 1."

Second, multiply the distance of center of gravity of the lever from the fulcrum by the weight of the lever and call the product "Product No. 2."

Third, multiply the distance the weight is from the fulcrum by the number of pounds contained in the weight and call the product "Product No. 3."

Fourth, add Products Nos. 1, 2 and 3 together and call the answer "The Sum."

Fifth, multiply the given pressure per square inch by the distance of center of valve from fulcrum, then divide "The Sum" by the product and call the quotient "Quotient No. 1."

Sixth, divide "Quotient No. 1" by .7854 and extract the square root of the last quotient and the answer will give the diameter, in inches, of the valve required.

TO DETERMINE THE DIAMETER OF THE VALVE REQUIRED IN THE DIAGRAM (FIG. 39), TO ALLOW THE VALVE TO RISE AT A PRESSURE OF 50 POUNDS PER SQUARE INCH.

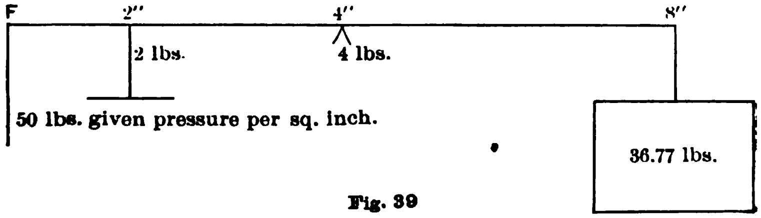


Fig. 39

F represents the fulcrum.

2″ represents the distance of the center of the valve from the fulcrum.

4″ represents the distance of the center of gravity of the lever from the fulcrum.

8″ represents the distance of the weight from the fulcrum.

2 lbs. represents the weight of the valve and spindle.

50 lbs. represents the pressure per square inch required to be carried.

FIRST OPERATION.

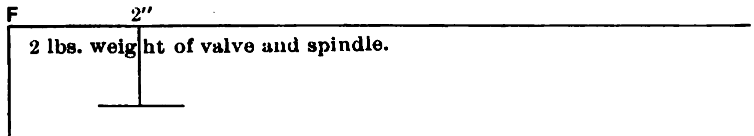


Fig. 40

RULE.—Multiply the distance of the center of the valve from the fulcrum by the weight of the valve and spindle and call the product "Product No. 1."

Example.—Let 2 lbs. equal weight of valve and spindle.

Let 2″ equal distance of center of valve from fulcrum.

Then we have: $2 \times 2 = 4$, " Product No. 1."

Or, which is the same thing:

2″ Distance of center of valve from fulcrum.
2 Weight of valve and spindle.
—
4 " Product No. 1."

SECOND OPERATION.

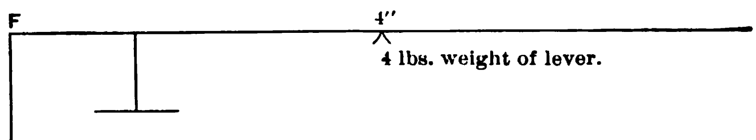


Fig. 41

RULE.—Multiply the distance of the center of gravity of the lever from the fulcrum by the weight of the lever and call the product " Product No. **2.**"

Example.—Let 4 lbs. equal weight of lever.

Let 4″ equal distance of center of gravity of lever from the fulcrum.

Then we have: $4 \times 4 = 16$, " Product No. 2."

Or, putting it in the ordinary form:

4″ Distance of center of gravity from fulcrum.
4 Weight of lever.
—
16 " Product No. 2."

THIRD OPERATION.

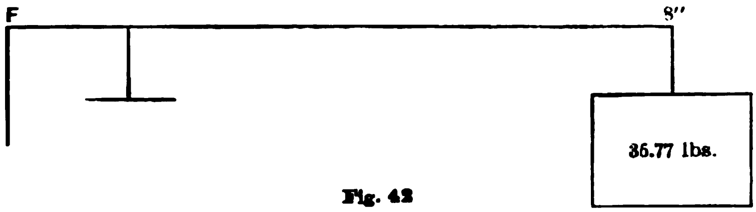


Fig. 42

RULE.—Multiply the number of pounds contained in the weight by the distance of the weight from the fulcrum and call the product "Product No. 3."

Example.—Let 36.77 lbs. equal weight of the weight.
Let 8″ equal distance of the weight from the fulcrum.

Then we have: $36.77 \times 8 = 294.16$, "Product No. 3."

Putting it in the ordinary form, we have:

36.77	Weight of the weight.
8″	Distance of weight from fulcrum.
294.16	"Product No. 3"

FOURTH OPERATION.

RULE.—Add Products Nos. 1, 2 *and* 3 *together and call the answer "The Sum."*

Example.—Let 4 equal "Product No. 1."
Let 16 equal "Product No. 2."
Let 294.16 equal "Product No. 3."

Then we have: $4+16+294.16=314.16$, "The Sum."

Putting it in the ordinary form, we have:

294.16	"Product No. 3."
16	"Product No. 2."
4	"Product No. 1."
314.16	"The Sum."

FIFTH OPERATION.

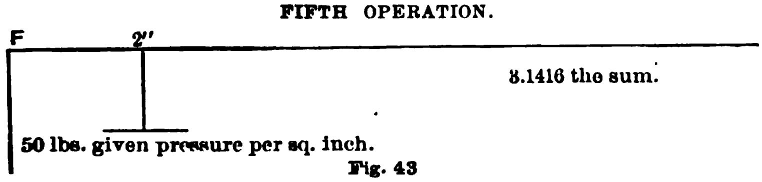


Fig. 43

RULE.—Multiply the given pressure by the distance of the center of the valve from the fulcrum, and divide "The Sum" by the product and call the quotient "Quotient No. 1."

Example.—Let 2″ equal the distance of center of valve from the fulcrum.

Let 50 lbs. equal the given pressure per square inch.

Let 314.16 equal "The Sum."

Then we have: $314.16 \div (50 \times 2) = 3.1416$, "Quotient No. 1."

In the ordinary form, we have:

```
50×2=100) 314.16 (3.1416  "Quotient No. 1."
          300
          ---
           141
           100
           ---
            416
            400
            ---
             160
             100
             ---
              600
              600
              ---
```

SIXTH OPERATION.

RULE.—Divide "Quotient No. 1" by .7854 and then extract the square root of the last quotient, the answer will give the diameter, in inches, of the required valve.

Example.—Let 3.1416 equal "Quotient No. 1."

Let .7854 equal a constant.

Then we have: $\sqrt{3.1416 \div .7854} = 2$ inches. Diameter of the required valve.

Putting it in the ordinary form, we have:

```
.7854) 3.1416 (4  Square of the diameter of the valve.
       3.1416
       ------
```

We now extract the square root of the quotient.

```
Thus:    4 (2 inches.  Diameter of required valve.
         4
        ---
```

Putting the foregoing operations in condensed form:

Example.—Let 2 inches equal distance of center of valve from the fulcrum.

Let 2 pounds equal estimated weight of valve and spindle.

Let 4 pounds equal weight of the lever.

Let 4 inches equal distance of center of gravity of lever from the fulcrum.

36.77 pounds equal weight of the weight.
8 inches equal distance of center of weight from fulcrum.
50 pounds equal given pressure per square inch.
.7854 equal a constant.

Then we have:

$$\sqrt{\frac{(2\times2)+(4\times4)+(36.77\times8)\div(50\times2)}{.7854}}=2 \text{ inches.}$$

Performing the operation, we have:

First, the weight of the valve and spindle multiplied by the distance of the center of the valve from the fulcrum. Thus: 2×2 equal "Product No. 1." 4

Second, the weight of the lever multiplied by the distance of the center of gravity of the lever from the fulcrum. Thus: 4×4 equal "Product No. 2." 16

Third, the weight of the weight multiplied by the distance of the center of the weight from the fulcrum. Thus: 36.77 $\times8$ equal "Product No. 3." 294.16

Fourth, Products Nos. 1, 2 and 3 added together:

Thus:

4	"Product No. 1."
16	"Product No. 2."
294.16	"Product No. 3."
314.16	"The Sum."

Fifth, the given pressure per square inch, multiplied by the distance of the center of the valve from the fulcrum, then "The Sum" divided by the product. Thus:

```
 50
  2
 ---   "The Sum."
100) 314.16 (3.1416  "Quotient No. 1."
     300
     ---
      141
      100
      ---
       416
       400
       ---
        160
        100
        ---
         600
         600
         ---
```

Or, in this manner: $\dfrac{314.16 \text{ "The Sum."}}{50\times2}=3.1416$ "Quotient No. 1."

Sixth, "Quotient No. 1" divided by .7854, and the square root extracted of the last quotient.

Thus: $\sqrt{3.1416 \div .7854} = 2$ inches. Diameter of valve.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} .7854)\,3.1416\,(4 \\ \underline{3.1416} \end{array}$$

And the square root of the quotient is:

$$\begin{array}{l} 4\,(2 \text{ inches. Diameter of the required valve.} \\ \underline{4} \end{array}$$

TO DETERMINE THE PRESSURE, PER SQUARE INCH, REQUIRED TO RAISE A VALVE WITHOUT WEIGHING ANY OF ITS PARTS.

RULE.—First, multiply the number of cubic inches of wrought-iron in the lever by .2817, *the product will give the number of pounds contained in the lever.*

Second, multiply the number of cubic inches of cast-iron in the weight by .2607, *and the product will give the number of pounds contained in the weight*

Third, square the diameter of the spindle, and multiply the product by .7854 *and multiply the second product by the length of the spindle, then multiply the last product by* .2817, *the answer will give the weight of the spindle in pounds.*

Fourth, immerse the valve in a small vessel of water, and from the rise of water in the vessel determine the number of cubic inches of water displaced by the valve, then if the valve is of brass, multiply the cubic inches of water displaced by .3194, *and if it is of cast-iron multiply by* .2607.

DETERMINE THE PRESSURE PER SQUARE INCH REQUIRED TO RAISE THE VALVE SHOWN IN FIG. 44.

FIRST OPERATION.

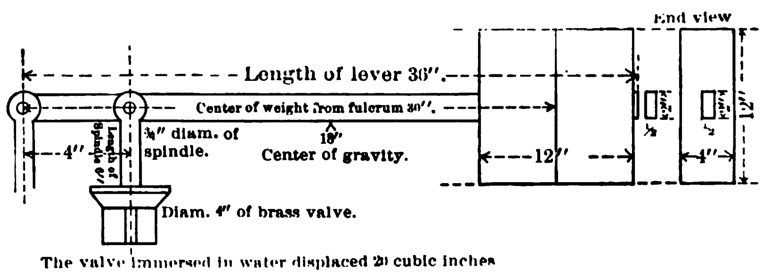


Fig. 44

RULE.—Multiply the length of the lever, in inches, by the width, then multiply the product by the thickness, then multiply the last product by .2817, *the answer will give the weight of the lever in pounds.*

Example.—Let 36″ equal length of the lever.
Let 2″ equal width of the lever.
Let 5 tenths of an inch equal thickness of the lever.
Let .2817 of a pound equal weight of one cubic inch of wrought-iron.

Then we have: $36\times2\times.5\times.2817=10.1412$ lbs. Weight of lever.

Putting it in the ordinary form, we have:

$$\begin{array}{r} 36 \\ 2 \\ \hline 72 \\ .5 \\ \hline 36.0 \\ .2817 \\ \hline 1\,6902 \\ 8\,451 \\ \hline 10.1412 \end{array}$$
lbs. Weight of lever.

This we will transfer to the following diagram, and continue until we get a complete diagram.

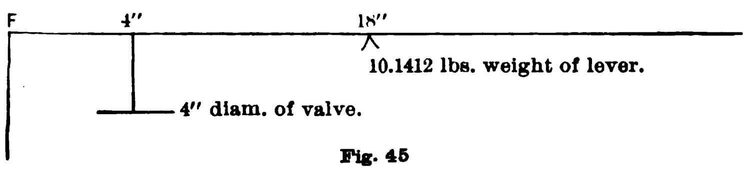


Fig. 45

SECOND OPERATION.

RULE.—Multiply the length of the cast-iron weight by its breadth, then multiply the product by the thickness of the weight, then deduct the area of the hole in the weight from the last product, then multiply the remainder by .2607, the answer will give the number of pounds contained in the weight

Example.—Let 12″ equal length of weight.
Let 12″ equal depth of weight.
Let 4″ equal thickness of weight.
Let 12″ equal length of hole in the weight.
Let 2″ equal height of hole in the weight.
Let 5 tenths of an inch equal width of hole in the weight.
Let .2607 of a pound equal weight of a cubic inch of cast-iron.

Then we have: $\overline{(12\times12\times4)-(12\times2\times.5)}\times.2607=147.0348$ lbs. Weight of weight.

6

Putting it in the ordinary form, we have:

$12 \times 12 \times 4 = 576$

$12 \times 2 \times .5 = 12$ Area of hole in the weight.

564 Number of cubic inches of iron in the weight.
.2607

1 0428
15 642
130 35

147.0348 lbs. Weight of weight.

We now have the weight of the lever and weight, which we transfer to the diagram (Fig.46).

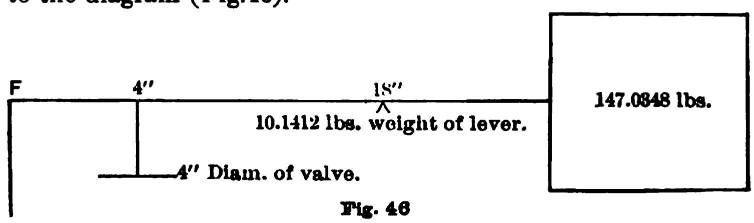


Fig. 46

THIRD OPERATION.

RULE.—Square the diameter of the spindle, then multiply the square of the diameter by .7854, *then multiply the product by the length of the spindle, then multiply the last product by* .2817, *and the answer will give the weight of the spindle in pounds.*

Example.—Let 75 one hundredths of an inch equal diameter of spindle.
Let .7854 equal a constant.
Let 6 inches equal length of spindle.
Let .2817 equal a constant.

Then we have: $.75^2 \times .7854 \times 6 \times .2817 = .7467092325$ lbs. Which is nearly $\frac{3}{4}$ of one pound.

Putting it in the ordinary way, we have:

.75
.75

375
525

.5625
.7854

22500
28125
45000
39375

Am't carried forward, .44178750

```
Am't brought forward,  .44178750
                               6
                      ----------
                      2.65072500
                           .2817
                      ----------
                        18555075
                        2650725
                      21205800
                      5301450
                      ----------
                    .7467092325 lbs.
```

Very nearly ¾ of one pound, which for convenience we will call it ¾ of a pound, or, .75 lbs.

We now have the weight of the lever, weight of weight, and weight of the spindle, which we transfer to the following diagram:

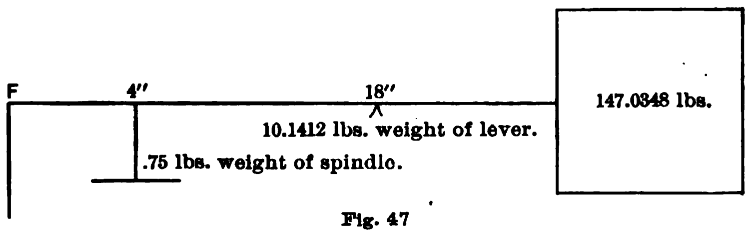


Fig. 47

FOURTH OPERATION.

RULE.—Multiply the diameter of the vessel of water by its diameter, and multiply the product by .7854, *then multiply the product by the rise of water, in inches, caused by immersion of the valve, then multiply the last product by* .3194 *if the valve is of brass, and by* .2607 *if the valve is of cast-iron.*

Example.—Let 4.25 inches equal diameter of vessel of water.
Let .7854 equal a constant.
Let 1.41 inches equal the rise of water in the vessel.
Let .3194 equal a constant for a brass valve.

Then we have: $4.25 \times 4.25 \times .7854 \times 1.41 \times .3194 = 6.38 +$ lbs. Weight of valve.

Putting it in the ordinary form, we have:

```
                         4.25
                         4.25
                        -----
                         2125
                         850
                       17 00
                       -------
Am't carried forward,  18.0625
```

Am't brought forward, 18.0625
.7854

722500
903125
1 445000
12 64375

14.18628750
1.41

1418628750
5 674515000
14 18628750

20.0026653750
.3194

800106615000
1800239883750
200026653750
6 00079961250

6.38885132077500 lbs. **Weight of valve.**

As it is unnecessary to carry the operation farther than two decimals, all the decimals except two will be dropped, because even the third decimal represents but 8 / 1000 of a pound.

We now have ascertained the weight of the lever, weight of the weight, weight of the spindle and weight of the valve, which we will transfer to the diagram (Fig. 48), and complete it for the purpose of determining the pressure per square inch required to raise it.

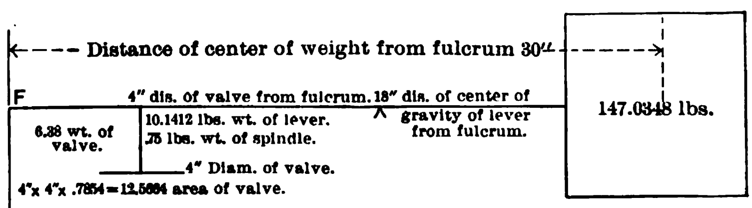


Fig. 48

FIRST OPERATION.

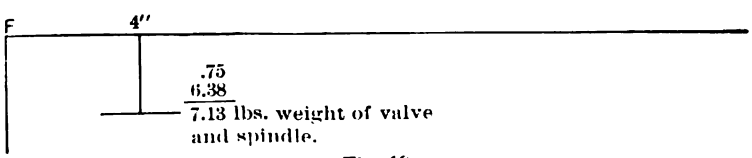


Fig. 49

RULE.—Multiply the weight of the valve and spindle by the distance the center of the valve is from the fulcrum, and call this product "Product No. 1."

Example.—Let 7.13 lbs. equal weight of valve and spindle.
Let 4″ equal distance of center of valve from fulcrum.

Then we have: $7.13 \times 4 = 28.52$, "Product No. 1."

Putting it in the ordinary form, we have.

$$\begin{array}{r} 7.13 \\ 4 \\ \hline 28.52 \end{array}$$ "Product No. 1."

SECOND OPERATION.

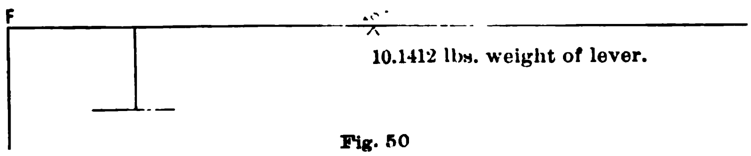


Fig. 50

RULE.—Multiply the weight of the lever by the distance of the center of gravity of the lever from the fulcrum, and call this product "Product No. 2."

Example.—Let 10.1412 lbs. equal weight of lever.
Let 18″ equal distance of center of gravity of lever from the fulcrum.

Then we have: $10.1412 \times 18 = 182.5416$, "Product No. 2."

Putting it in the ordinary way, we have:

$$\begin{array}{r} 10.1412 \\ 18 \\ \hline 81\ 1296 \\ 101\ 412 \\ \hline 182.5416 \end{array}$$ "Product No. 2."

THIRD OPERATION.

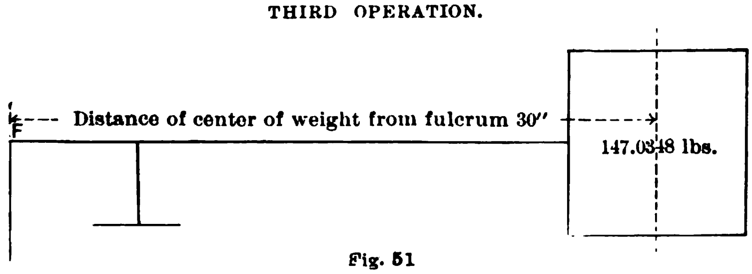


Fig. 51

RULE.—Multiply the weight of the weight by the distance its center is from the fulcrum, and call the product "Product No. 3."

Example.—Let 147.0348 lbs. equal weight of the weight.
Let 30″ equal distance of center of weight from the fulcrum.

Then we have: $147.0348 \times 30 = 4411.044$, "Product No. 3."

Putting it in the ordinary form, we have:

$$\begin{array}{r} 147.0348 \\ 30 \\ \hline 4411.0440 \end{array} \quad \text{"Product No. 3."}$$

FOURTH OPERATION.

RULE.—Add Products Nos. 1, 2, *and* 3, *together and call the answer "The Sum."*

Example.—Let 28.52 equal "Product No. 1."
Let 182.5416 equal "Product No. 2."
Let 4411.0440 equal "Product No. 3."

Then we have: $28.52 + 182.5416 + 4411.0440 = 4622.1056$, "The Sum."

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl} 28.52 & \text{"Product No. 1."} \\ 182.5416 & \text{"Product No. 2."} \\ 4411.0440 & \text{"Product No. 3."} \\ \hline 4622.1056 & \text{"The Sum."} \end{array}$$

FIFTH OPERATION.

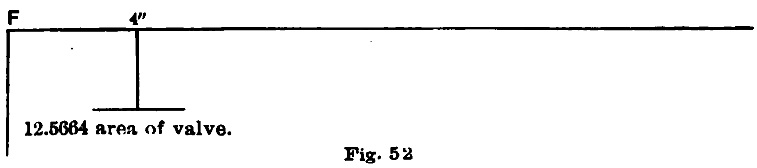


Fig. 52

RULE.—Multiply the area of the valve by the distance of its center from the fulcrum, and call the product "Product No. 4."

Example.—Let 12.5664 square inches equal area of valve.
Let 4 inches equal distance of center of valve from the fulcrum.

Then we have: $12.5664 \times 4 = 50.2656$, "Product No. 4."

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 12.5664 \\ 4 \\ \hline 50.2656 \end{array} \quad \text{"Product No. 4."}$$

SIXTH OPERATION.

RULE.—Divide "The Sum" by "Product No. 4," and the quotient will give the pressure per square inch required to raise the valve as adjusted in Fig. 48.

Example.—Let 4622.1056 equal, "The Sum."
Let 50.2656 equal, "Product No. 4."

Then we have: $4622.1056 \div 50.2656 = 91.95+$ lbs. Pressure per square inch required to raise the valve.

Performing the operation, we have:

```
50.2656) 4622.1056 (91.95+lbs.   Pressure per square inch
         4523 904                required to raise the
         --------                valve.
           98 2016
           50 2656
           -------
           47 93600
           45 23904
           --------
            2 696960
            2 513280
            --------
```

SIMPLE METHOD OF ADJUSTING SAFETY-VALVES.

Let it be required to determine the pressure necessary to raise a valve adjusted according to the following diagram:

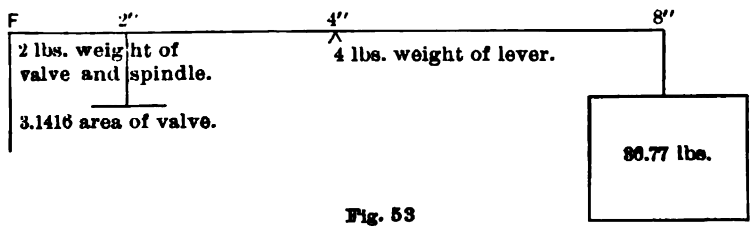


Fig. 53

RULE.—Take a bar of iron, of uniform section, at least twice the length of the lever of the safety-valve, balance the bar on the edge of a cold-chisel in a vise, and mark the point on the bar at which it balances, and call that "The Fulcrum." Then, to the right of that point, put another mark the same distance the valve is from the fulcrum; then balance the lever of the safety-valve, then measure the distance from the fulcrum to the point at which the lever balances and place a mark the same distance on the bar to the right of the point on which it balances, then measure the distance the weight is to be placed on the lever from the fulcrum, and lay off the same distance on the bar to the right of the point on which the bar balances; then lay off a distance to the left of the point on which the bar balances equal to the distance the valve is from the fulcrum. Then place the bar on the cold-chisel in the vise on the point at which it balances, and hang the valve and spindle, to the right on the bar from the point on which the bar balances, the same distance the valve is from the fulcrum when the valve is in its position; then hang the lever of the valve on the bar to the right of the point on which the bar balances, the same distance that the lever balances from its fulcrum; then hang the weight of the

safety-valve on the bar to the right of the point on which the bar balances the same distance the weight is to be placed on the lever of the valve; then hang weights on the opposite side of the bar, to the left of the point on which the bar balances, the same distance the valve is from the fulcrum, and continue to add weight at that point until the lever balances the valve, spindle, lever and weight on the opposite side of the bar; then remove and weigh the weights hung on the left half of the bar; then divide the number of pounds contained in the weights by the area of the valve and the quotient will give the number of pounds pressure per square inch required to raise the valve.

We will now proceed to arrange the points mentioned in the rule, on a bar of iron (as shown in Fig. 54), preparatory to hanging the different parts of the safety-valve (in Fig. 53) on it.

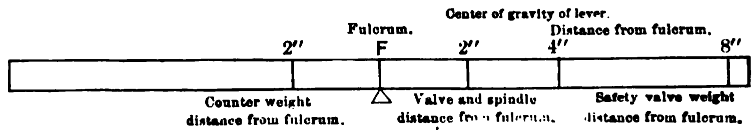


Fig. 54

We will now construct a diagram (Fig. 55), showing the counter-weight, and the different parts of the safety-valve (as shown in Fig. 53) hung in position.

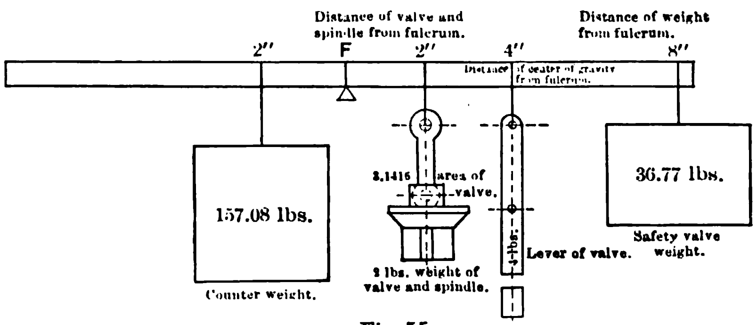


Fig. 55

It will be observed that each part of the safety-valve has the weight given in the diagram; that part of the diagram is entirely unnecessary to enable the student to determine the pressure per square inch required to raise the valve, and we only place them in the diagram for the purpose of making a calculation hereafter to prove the correctness of this method of determining the pressure required to raise the valve. We will now proceed to explain the *modus operandi* in detail.

First, ascertain the distance the valve and spindle are from the fulcrum, then hang them at the same distance to the right of F on the bar.

Second, ascertain how far from the fulcrum the safety-valve lever will balance on a knife-edge, and hang the lever at the same distance to the right of F on the bar.

Third, ascertain the distance the weight of the safety-valve is to be placed on the lever from the fulcrum, and hang it the same distance to the right of F on the bar.

Fourth, hang weights to the left of F on the bar the same distance the valve and spindle hang to the right of F; and continue to add weight until the bar balances, and call these "the counter-weights."

Fifth, remove the counter-weights and weigh them, then divide the weight of the counter-weights by the area of the valve, the quotient will give the pressure, per square inch, required to raise the valve.

Example.—Let 157.08 pounds equal weight of counter-weights.
Let 3.1416 square inches equal area of the valve.

Then we have: $157.08 \div 3.1416 = 50$ lbs. Pressure required to raise the valve.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 3.1416)\ 157.0800\ (50 \text{ lbs.} \\ \underline{157.080}\phantom{\ (50 \text{ lbs.}} \\ 0\phantom{\ (50 \text{ lbs.}} \end{array}$$

Pressure required to raise the valve, shown in Fig. 53.

NOTE. In performing division with decimals, both the divisor and dividend should contain the same number of decimals; if either contains less than the other a sufficient number of ciphers should be annexed to the lesser to make the number of decimals in that equal to the number of decimals in the greater; for that reason, it will be observed, two ciphers have been added to the decimals in the dividend in the above example.

The student will understand that there are two forces which operate on a safety-valve, the one is downward and the other is upward. The downward pressure on the valve seat is produced by three forces: the weight of the valve and spindle, the pressure produced on the valve seat by the weight of the lever, and the pressure produced on the valve seat by the weight of the weight. The upward force consists of the pressure against the bottom of the valve, if this is greater than the combined downward forces the valve will rise, if it is less the valve will not rise. The student, therefore, must not get these forces confounded with any pressure, strain or weight at the fulcrum, but his attention must be directed to the result produced at the end of the short arm of the lever at the center of the valve. In Fig. 55, the downward forces consist of the weights of the valve, spindle, lever and weight, and the upward force consists of the counter-weight, because it has a tendency to raise the valve, hence it is the same as though a total steam pressure of 157.08 pounds was under the valve.

TO DETERMINE THE ACTUAL PRESSURE PRODUCED ON THE VALVE SEAT BY THE VALVE, SPINDLE, LEVER AND WEIGHT.

RULE.—Divide the distance of the lever from the fulcrum by the distance the valve is from the fulcrum, then multiply the weight of the lever by the quotient, the answer will give the total pressure in pounds produced downward at the valve by the weight of the lever.

Second, divide the distance the weight is from the fulcrum by the distance the valve is from the fulcrum, then multiply the weight of the weight by the quotient, and the answer will give the total pressure produced downward at the valve by the weight of the weight.

Third, as the valve and spindle are directly under the end of the short arm of the lever, the pressure produced downward at that point by the valve and spindle is equal to their combined weight, and requires no calculation.

Fourth, add the pressure produced downward at the valve by the weight of the valve, spindle, lever and weight, and the sum will give the total pressure produced downward at the end of the short arm of the lever, or the point at which the valve and spindle are hung.

Example.—Let 2 inches equal distance of center of valve from the fulcrum.
Let 2 pounds equal weight of valve and spindle.
Let 4 inches equal distance of lever from the fulcrum.
Let 4 pounds equal weight of lever.
Let 8 inches equal distance of weight from the fulcrum.
Let 36.77 pounds equal weight of the weight.

Then we have:

First, the weight of the valve and spindle on the valve seat equal: 2 lbs.

Second, the distance of the lever from the fulcrum divided by the distance the center of the valve is from the fulcrum, and the weight of the lever multiplied by the quotient: $\frac{4}{2}\times 4=$ 8 lbs.

Third, the distance of the weight from the fulcrum divided by the distance the center of the valve is from the fulcrum, and the weight of the weight multiplied by the quotient: $\frac{8}{2}\times 36.77=$ 147.08 lbs.

Fourth, the pressure produced downward at the valve by the weight of the valve, spindle, lever and weight added together: 157.08 lbs.

It will be observed that in Fig. 55, the total pressure produced on the valve seat by the weight of the valve, spindle, lever and weight is 157.08 pounds, and that this pressure is downward two inches from the fulcrum to the right. It will also be observed that the weight 157.08 pounds, two inches to the left of the fulcrum, while its strain is downward at that point, produces a strain upward of 157.08 pounds, two inches to the right of the fulcrum, and hence it exactly balances the forces to the right of the fulcrum. We will now suppose the strain upward two inches from the fulcrum to the right, produced by the counter-weight to the left to be a total strain pressure of 157.08 pounds under the valve, by dividing that total strain by the area of the valve, we will have the pressure per square inch required to raise the valve.

Example.—Let 157.08 pounds equal total pressure under the valve.
Let 3.1416 square inches equal area of the valve.

Then we have: $157.08 \div 3.1416 = 50$ lbs. per square inch.

Now, as the 157.08 pounds total strain under the valve, is exactly counter-balanced by the weight of the valve, spindle, lever and weight, any increase in the total strain will raise the valve.

HOW TO GRADUATE A SAFETY-VALVE LEVER.

The method shown in Fig. 55 is a simple as well as an accurate method for graduating the lever of a safety-valve, so that the weight can readily be adjusted to allow the valve to rise at any desired pressure, per square inch, and also to test the accuracy of the steam gauge. We will, therefore, proceed to show how to graduate a safety-valve lever by the use of the bar, as shown in Fig. 55.

RULE.—First, multiply the desired pressure, per square inch, by the area of the valve, and the product will give the number of pounds of weight to be hung at the point where the counter-weight is in Fig. 55. In other words, hang the weight on the bar to the left of the fulcrum the same distance that the valve is from the fulcrum of the lever to be graduated.

Second, move the weight of the safety-valve on the bar to a point where it will balance the counter-weight on the left of the fulcrum, and that will be the point at which the weight must be placed on the lever when the valve is in position on the boiler, to allow the valve to rise at the desired pressure.

Third, mark the distance from the fulcrum on the lever of the valve at which the weight balanced the bar from the fulcrum of the bar, and mark the pressure, per square inch, at that point at which the valve will rise. Continue these operations for all desired pressures.

Fourth, while performing the operation, the valve, spindle and lever must be kept constantly in their respective positions, as shown in Fig. 55.

TO DETERMINE THE PROPER AREA OF SAFETY-VALVES.

The proper area of a safety-valve must be determined by the capacity of the boiler for generating steam, to which it is attached. In all cases, however, the area of the valve should be sufficient, under all circumstances, to relieve the boiler automatically of excess of pressure, and keep the pressure within the limit of perfect safety. In other words, all safety-valves should have an area large enough to prevent the accumulation of pressure in the boiler beyond that at which the valve is set to blow off.

The size of the valve required depends in a great measure upon its construction; because the area of opening produced by the lift of the valve under steam pressure is the measure of the valve's efficiency. There are valves that will discharge double the amount of steam that other valves will of the same diameter. Great care should, therefore, be exercised in the selection of safety-valves, and under no circumstances should any valve be selected that is in any way faulty in construction. And as to the size of the valve required, when selecting a lever-valve, scientific experiments and every-day practice have determined that a lever-valve, to do its work effectively, should have an area of one square inch for every two square feet of the grate surface of the boiler to which it is to be attached; and it might as well be emphatically understood that no valve is a safety-valve unless it will, automatically, relieve the boiler and prevent the pressure of steam from crossing the danger line in the strength of the boiler.

TO DETERMINE THE PROPER DIAMETER OF A SAFETY-VALVE.

RULE.—First, divide the number of square feet of grate surface of the boiler by 2, *and call the quotient "Quotient No.* 1."

Second, divide " Quotient No. 1 " *by* .7854 *and extract the square root of the last quotient, the answer will give the diameter of the valve.*

Example.—Let 20 square feet equal grate surface of the boiler.
Let 2 equal a constant.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{20 \div 2}{.7854}} = 3.56+ \text{ inches. } \textbf{Diameter of valve required.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 2)\ 20 \\ \hline .7854)\ 10.0000\ (12.7323+ \\ 7\ 854 \\ \hline 2\ 1460 \end{array}$$

Continued on next page,

Brought from previous page,

```
    2 1460
    1 5708
    ------
      57520
      54978
      -----
       25420
       23562
       -----
        18580
        15708
        -----
         28720
         23562
         -----
```

We will now proceed to extract the square root of the last quotient. Thus:

```
       . . .
      12.7323 (3.56 + inches.  Diameter of valve required.
       9
      ---
  65) 373
      325
      ----
 706) 4823
      3246
      ----
```

Another method, and one which will produce the same result is as follows:

RULE.—Divide the number of square feet in the grate surface by the constant 1.5708, *and extract the square root of the quotient, the answer will give the diameter of the valve.*

Example.—Taking the same number of square feet of grate surface taken in the preceding example.

We have: $\sqrt{20 \div 1.5708} = 3.56+$ inches. Diameter of valve.

Performing the operation in the ordinary way, we have:

```
1.5708) 20.0000 (12.7323+
        15 708
        ------
         4 2920
         3 1416
         ------
         1 15040
         1 09956
         -------
            50840
            47124
            -----
             37160
             31416
             -----
              57440
              47124
              -----
```

Extracting the square root of the quotient, we have:

```
   . . .
   12.7323 (3.56+ inches.   Diameter of valve.
    9
   -------
 65) 3 73
     3 25
     -----
 706) 4823
      4236
      ----
```

CHAPTER III.

VALVES GENERALLY.

AREA OF OPENING—FLAT-SEATED VALVES.

TO DETERMINE THE AREA OF OPENING PRODUCED BY THE LIFT OF A FLAT-SEATED VALVE.

RULE.—Multiply the diameter of the valve by 3.1416, *and then multiply the product by the lift of the valve, in hundredths of an inch, the answer will give the area of opening produced by the lift of the valve.*

Example.—Let 4 inches equal diameter of the valve.
Let 3.1416 equal a constant.
Let 25 one hundredths of an inch equal lift of the valve.

Then we have: $4 \times 3.1416 \times .25 = 3.1416$ inches. Area of opening produced by the lift of the valve.

The 3.1416 inches represent an opening one inch in width, and 3.1416 inches in length.

TO DETERMINE THE SIZE OF A SQUARE HOLE FROM THE AREA OF OPENING PRODUCED BY THE LIFT OF A FLAT-SEATED VALVE.

RULE.—Extract the square root of the quotient in the above example, the answer will give the size of the hole in square form.

Example.—Let 3.1416 equal number of square inches in area of valve opening.

Then we have.

$$
\begin{array}{rl}
 & \dot{3}.1\dot{4}1\dot{6}\,(1.77+ \text{ inches square.} \\
 & 1 \\
27\,) & \overline{2}14 \\
 & 189 \\
347\,) & \overline{2516} \\
 & 2429 \\
 & \overline{}
\end{array}
$$

The size hole the area of opening produced by the lift of the valve.

TO DETERMINE THE SIZE OF A ROUND HOLE FROM THE AREA OF OPENING PRODUCED BY THE LIFT OF A FLAT-SEATED VALVE.

RULE.—Divide the area of opening produced by the lift of the valve by .7854, *and then extract the square root of the quotient, the answer will give the diameter of the hole.*

Example.—Let 3.1416 equal area of opening produced by the lift of the valve.

Let .7854 equal a constant.

Then we have: $\sqrt{3.1416 \div .7854} = 2$ inches.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} .7854\,)\,3.1416\,(\,4 \\ 3.1416 \\ \hline \end{array}$$

Extracting the square root of the quotient, we have:

$$\begin{array}{l} \dot{4}\,(\,2 \text{ inches.} \\ 4 \\ \hline \end{array}$$ Diameter of round hole 3.1416 square inches will make.

AREA OF OPENING—BEVEL-SEATED SAFETY-VALVES.

The rule for determining the area of the opening produced by the lift of bevel-seated valves differs materially from that for flat-seated valves. The reason is that as the angle of the seat deviates from that of 90 degrees to the center line of the valve's axis, the opening produced by the lift becomes less in proportion to the lift, as the angle approaches a line parallel to the center line of the valve's axis.

A flat-seated valve lifting a distance equal to one-quarter of its diameter, produces an area of opening equal to the area of the diameter of the valve; while a bevel-seated valve, until it rises above its seat, produces an area of opening less in proportion as the angle of the seat deviates from an angle of 90 degrees to the center line of the valve's axis. In other words, taking a flat-seated valve the opening produced by its lift is equal to the lift, while that of a bevel-seated valve is less than the lift in proportion as the bevel becomes greater. It will be observed, that as the angle of a bevel-seated valve approaches a perpendicular line, the opening produced by the lift becomes less in proportion. The bevel might be made so great that the seat of the valve would become very nearly a cylinder, in which event the valve would have to lift above the seat before it would produce any perceptible opening; and as the area of opening produced by the lift of the valve is governed by the opening produced by the lift, it will be seen that the greater the angle of the seat the less opening any given lift will produce. Take, for example, a flat-seated valve and a bevel-seated valve, each four inches in diameter, and the bevel-seated valve having its seat one inch in depth, and made at an angle of 45 degrees to the center line of the valve's axis, and each lifting one inch; the area of opening produced by the lift of the flat-seated valve would be 12.5664 square inches, while that of the bevel-seated valve would be but 10.4547+ square inches. We will now proceed to give the rule relating to bevel-seated valves.

TO DETERMINE THE AREA OF OPENING PRODUCED BY THE LIFT OF ANY BEVEL-SEATED VALVE WHEN THE LIFT IS NOT ABOVE THE SEAT.

RULE.—First, multiply the diameter of the valve in inches by the lift in hundredths of an inch, and then multiply the product by the sine of the angle of the seat, and call the last product "Product No. 1."

Second, multiply the square of the lift in hundredths of an inch by the square of the sine of the angle of the seat, then multiply the product by the cosine of the angle of the seat, and call the last product "Product No. 2."

Third, add Products Nos. 1 and 2 together and multiply the sum of the two by 3.1416 the product will give the area of opening produced by the lift of the valve.

Example.—Let D equal diameter of the valve in inches.
Let l equal lift of the valve in hundredths of an inch.
Let s equal sine of the angle of the seat.
Let c equal cosine of the angle of the seat.
Let C equal 3.1416 a constant.
A equal area of opening produced by the lift of the valve.

Then we have: $[\,(D\times l\times s)+(l^2\times s^2\times c)\,]\times C=A.$

Substituting figures for the values in the formula:

Let 4 inches equal diameter of the valve.
Let 12 one hundredths of an inch equal lift of valve.
Let .707 equal sine of an angle of 45 degrees (bevel of seat).
Let .707 equal cosine of an angle of 45 degrees (bevel of seat).
Let 3.1416 equal a constant.

Then we have:

$\overline{(4\times .12\times .707)+(.12^2\times .707^2\times .707)}\times 3.1416=1.08+$ square inches. Area of opening produced by the lift.

Performing the operation in the ordinary way, we have:

$4\times .12\times .707=.33936$ "Product No. 1."
$.12\times .12\times .707\times .707\times .707=.0050888626992$ "Product No. 2."

The sum multiplied by 3.1416, $.3444488626992\times 3.1416=$ 1.08212054705580672 square inches. Area of opening produced by lift of valve.

TO DETERMINE THE AREA OF OPENING PRODUCED BY THE LIFT OF A BEVEL-SEATED VALVE WHERE THE VALVE LIFTS ABOVE THE SEAT.

RULE.—First, multiply the diameter of the valve in inches by the depth of the seat, in hundredths of an inch, then multiply the product by the sine of the angle of the seat, and call the last product "Product No. 1."

Second, multiply the square of the depth of the seat in hundredths of an inch by the square of the sine of the angle of the seat, then multiply the product by the cosine of the angle of the seat, and call the last product "Product No. 2."

Third, multiply that part of the lift of the valve above the seat, in hundredths of an inch, by the diameter of the valve, in inches, and call the product "Product No. 3."

Fourth, add Products Nos. 1, 2 *and* 3 *together and call the answer " The Sum."*

Fifth, multiply " The Sum " by 3.1416 *and the product will give the area of opening, in square inches, produced by the lift of the valve.*

Example.—Let D equal diameter of the valve in inches.
Let d equal depth of the seat, in hundredths of an inch.
Let s equal sine of the angle of the seat.
Let c equal cosine of the angle of the seat.
Let l equal lift of valve in hundredths of an inch.
Let C equal 3.1416, a constant.
Let A equal area of opening produced by lift of valve.

Then we have: $[(D\times d\times s)+(d^2\times s^2\times c)+(D\times l-d)]\times C=A.$

To simplify the operation we will put it in the following form:

Taking a valve 2 inches in diameter and lifting .5 of an inch, with a depth of seat of .25 of an inch, and an angle of seat of 45 degrees.

Then we have:

$$D\times d\times s= 2\times .25\times .707 = .35350 \quad \text{" Product No. 1."}$$
$$d^2\times s^2\times c=.25^2\times .707^2\times .707 = .02208+ \quad \text{" Product No. 2."}$$
$$D\times(l-d) = 2\times(.50-.25) = .50 \quad \text{" Product No. 3."}$$
$$.87558 \quad \text{" The Sum."}$$

Then, multiplying the sum by 3.1416, we have:

```
       .87558
       3.1416
      -------
       525348
       87558
     350232
     87558
   2 62674
   ----------
   2.750722128 square inches.
```
Area of opening produced by the lift of the valve.

To simplify the operation still further we will put it in the following form:

Example.—Let 2 inches equal diameter of valve.

Let 25 one hundredths of an inch equal depth of seat of valve.

Let .707 equal sine of angle (45°) of seat.

Let .707 equal cosine of angle (45°) of seat.

Let 50 one hundredths of an inch equal lift of valve.

Let 3.1416 equal a constant.

Then we have:

$$\overline{(2\times.25\times.707)+(.25^2\times.707^2\times.707)+(2\times.50-.25)}\times3.1416=2.75+.$$

Performing the operation, we have:

.25″ Depth of seat.
2″ Diameter of valve.
.50
.707 Sine of the angle of the seat.
.35350 "Product No. 1."

Next we have:
.25″ Depth of seat.
.25″ Depth of seat.
125
50
.0625 Square of depth of seat.

Next we have:
.707 Sine of the angle of the seat.
.707 Sine of the angle of the seat.
4949
4949
Square of the sine of the angle of the seat. .499849
.0625 Square of depth of the seat.
2499245
999698
2999094
.0312405625
.707 Cosine of the angle of the seat.
2186839375
2186839375
.0220870776875 "Product No. 2."

Next we have:
.50″ Lift of valve.
.25″ Depth of seat subtracted from lift of valve.
.25 Lift of valve above the seat.
2 Diameter of valve.
.50 "Product No. 3."

Next, adding Products Nos. 1, 2 and 3 together, we have:

.35350	"Product No. 1."
.02208+	"Product No. 2."
.50	"Product No. 3."
.87558	"The Sum."

Finally, multiplying "The Sum" by the constant 3.1416, we have:

```
       .87558
       3.1416
      -------
       525348
       87558
     350232
     87558
   2 62674
   -----------
   2.750722128 square inches.
```

Area of opening produced by the lift of the valve.

SINES AND COSINES.

The following table contains the sines and cosines of the angles at which the seats of bevel-seated valves are commonly made; it is therefore given for convenient reference and the use of the student:

Angle.	Sine.	Cosine.	Angle.	Sine.	Cosine.
25°	.42262	.90631	38°	.61566	.78801
26°	.43837	.89879	39°	.62932	.77715
27°	.45399	.89101	40°	.64279	.76604
28°	.46947	.88295	41°	.65606	.75471
29°	.48481	.87462	42°	.66913	.74314
30°	.5	.86603	43°	.682	.73135
31°	.51504	.85717	44°	.69466	.71934
32°	.52992	.84805	45°	.70711	.70711
33°	.54464	.83867	46°	.71934	.69466
34°	.55919	.82904	47°	.73135	.682
35°	.57358	.81915	48°	.74314	.66913
36°	.58779	.80902	49°	.75471	.65606
37°	.60182	.79864	50°	.76604	.64279

As sines and cosines of angles enter largely into the science of steam and mechanical engineering, and particularly into that branch embraced in the steam boiler safety-valve, it is very important that the student should understand just what sines and cosines are. As the use of the diagram is the simplest method that can be employed, the attention of the student is directed to Fig. 56, and to a careful study of the same.

It will be observed that the line C B represents the valve seat, which is made at an angle of 45°. Dotted line A B and dotted line from C to D represent the cosine, and the dotted line B D the sine of the angle. The angle of the seat of the valve being 45° the lines representing the sines and cosines are exactly the same length. It will also be observed that as the line representing the valve seat leaves the dotted line C E and approaches the perpendicular dotted line C F that the cosines become shorter and the sine becomes longer; and the opening produced by the lift of the valve becomes correspondingly less; hence it is, as previously stated, that as the angle of the valve seat deviates from the angle of 90° from the center line of the valve's axis, the opening produced by the lift becomes less. The perpendicular dotted line C F represents the center line of the valve's axis, and the horizontal dotted line C E has an angle of 90° from the perpendicular line C F, the angle at which flat-seated valves are made, and this line is called the radius, or, rather the distance from C to E on this line is called the radius.

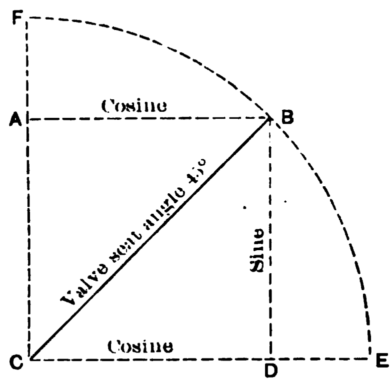


Fig. 56

By the use of the information thus obtained, and the employment of one of the simplest rules in trigonometry, we are enabled to formulate a very simple rule for determining the area of opening produced by the lift of a bevel-seated valve, having its seat made at an angle of 45° to the center line of the valve's axis. Practice and scientific experiments—both have demonstrated that 45° is the proper angle at which the seats of all safety-valves should be made, and that has been adopted as the standard by the United States and British governments. Therefore, the following rule, based upon the standard of an angle of 45°, will be found practically correct without the use of sines or cosines, beside the confusion produced in the mind of the student by the use of sines and cosines is obviated.

TO DETERMINE THE AREA OF OPENING PRODUCED BY THE LIFT OF A BEVEL-SEATED VALVE HAVING ITS SEAT MADE AT AN ANGLE OF 45° TO THE CENTER LINE OF THE VALVE'S AXIS WITHOUT THE USE OF SINES OR COSINES.

RULE.—First, add one-half of the lift of the valve, in hundredths of an inch, to the diameter of the valve, and multiply the sum by the constant 3.1416, *and call the product "Product No. 1."*

Second, multiply the lift by .707, and call the product "Product No. 2."

Third, multiply "Product No. 1" by "Product No. 2" and the product will give the area of opening produced by the lift of the valve.

Example.—Let 4 inches equal diameter of the valve.
Let 20 one hundredths of an inch equal lift of valve.
Let 3.1416 equal a constant.

Then we have

$$\left(4+\frac{.20}{2}\right)\times 3.1416\times .707\times .20=1.82+ \text{ square inches.}$$

Area of opening produced by the lift of the valve.

Performing the operation, we have:

2).20 inches. **Lift of valve.**
.10 inches. **One-half of lift of valve.**
4. inches. **Diameter of valve.**
4.10
3.1416
2460
410
1640
410
12 30
12.880560 **"Product No. 1."**

Next we have:

.707
.20 **Lift of valve.**
.14140 **"Product No. 2."**

Finally we have:

12.880560 **"Product No. 1."**
.14140 **"Product No. 2."**
515222400
12880560
51522240
1 2880560
1.82131118400 square inches. **Area of opening produced by the lift of the valve.**

The next thing will be to illustrate by diagram how the area of opening produced by the lift of the valve is obtained, and why one-half of the lift is added to the diameter of the valve in performing the operation. This is shown in Fig. 57, in which the seat is made purposely large for greater convenience in illustrating the subject:

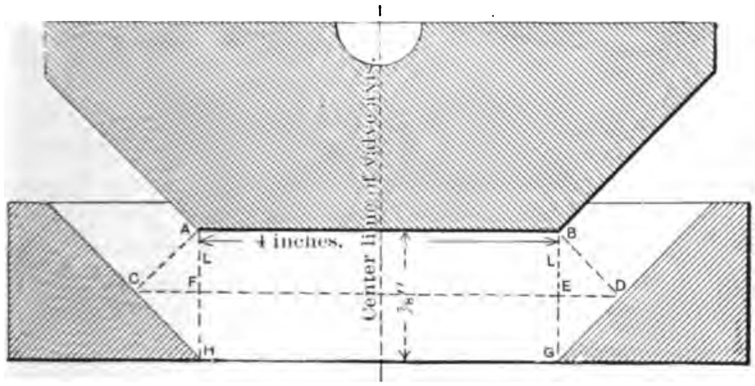


Fig. 57

It will be observed that the opening produced by the lift is conic shaped, and that it forms the frustrum of a cone, of which A C and B D form the slant height, A B the diameter of the top, and C D the diameter of the base, and the perpendicular dotted lines L L represent the lift of the valve.

If we add the diameter of the top to the diameter of the base and divide the sum by 2 we get the average diameter of the frustrum of the cone. Then, if we multiply the average diameter by 3.1416 we get the average circumference, then if we multiply the average circumference by the length of the slant side A C or B D we get the surface of the frustrum of the cone, and the surface represents the area of opening produced by the lift of the valve.

It will also be observed that A C H and B D G form right angle triangles of equal sides, and that the horizontal distance from C to F and E to D is the same as the perpendicular distance from F to A and E to B, or H to F and G to E. Therefore, if we add one-half of the lift to the diameter of the valve A B, we have the average diameter of the frustrum of the cone, and this is done simply to obviate the unnecessary labor of adding the diameter of the top of the cone to the diameter of the bottom and dividing the sum, because simply adding one-half of the lift of the valve to its diameter accomplishes the same result exactly, and has the advantage of being more simple and requiring much less labor.

To find the amount of opening the lift of the valve will produce, it must be observed that A H and B G each form the hypothenuse of a right angle triangle. Therefore, to get the length of the sides A C or B D we multiply the hypothenuse, or the lift of the valve which is the same thing in this case, by .707, and the product will give the length of the sides A C or B D, or in other words, it will give the opening produced by the lift of the valve.

To illustrate: suppose the valve (as shown in Fig. 57), was 4 inches in diameter and that its lift was $\frac{7}{8}$ of an inch, what is the area of opening produced by the lift.

Then, according to rule, we add one-half of the lift, $\frac{7}{8}$ of an inch, which, expressed in decimals is .875 and $.875 \div 2 = .4375$, and .4375 added to 4 inches (the diameter of the valve) will give an average diameter of 4.4375 inches, and this multiplied by 3.1416 will give the average circumference. Thus: $4.4375 \times 3.1416 = 13.94+$ inches, average circumference of the valve.

Performing the operation, we have:

```
      4.4375
      3.1416
     -------
      266250
     44375
   177500
   44375
 13 3125
 -----------
 13.94085000 inches.
```

Average circumference of the valve.

Then, if we multiply the average circumference of the valve by the length of the slant sides A C or B D of the conic shaped opening, we will get the area of opening, produced by the lift of the valve.

The length of the slant side of the conic shaped opening is obtained by multiplying the lift .875 by .707, thus: $.875 \times .707 = .618625$, decimals of an inch.

Performing the operation, we have:

```
    .875
    .707
    ----
    6125
  6125
 -------
 .618625
```

Decimals of an inch, the length of the side of the cone and the actual opening produced by the lift of the valve.

Now, if we multiply this opening by the average circumference we will then have: $.618625 \times 13.94 = 8.62363250$ square inches.

Performing the operation, we have:

```
        .618625
          13.94
      ---------
        2474500
       5567625
     1 855875
     6 18625
     ----------
     8.623632500 square inches.
```

Area of opening produced by the lift of the valve.

It will be observed that while the lift of the valve is .875 of an inch, the actual opening for the discharge of steam or liquid is from A to C or B to D, and that it amounts to .618625 of an inch—a fraction less than $\frac{5}{8}$ while the lift is $\frac{7}{8}$; and that the difference between the lift and the opening produced by it becomes greater as the line of the valve seat approaches the center line of the valve's axis perpendicularly.

VALVE PROPORTIONS.

It is very important that every engineer should understand not only how to proportion and adjust safety-valves, but he should understand how to proportion and adjust all valves connected with pipes employed to convey steam, water or other fluids, so as to offer as little resistance as possible. It frequently occurs that a valve is required to have too much lift for efficient performance of the work required of it. In such cases the valve should be replaced by one having an area sufficient to perform the functions required of it without the necessity of giving it a lift so great as to cause the valve to pound itself to pieces in a short time. It is therefore very essential that the engineer should understand how to determine the size of the valve required.

LIFT OF FLAT-SEATED VALVES.

TO DETERMINE THE LIFT OF A REQUIRED FLAT-SEATED VALVE TO PRODUCE AN AREA OF OPENING EQUAL TO THE AREA OF OPENING PRODUCED BY THE LIFT OF A GIVEN VALVE.

RULE.—Divide the lift of the given valve in hundredths of an inch, by the diameter of the required valve, and multiply the quotient by the diameter of the given valve.

Example.—Let 25 one hundredths of an inch equal lift of the given valve.

Let 4 inches equal diameter of the required valve.

Let 3 inches equal diameter of the given valve.

Then we have:

$$\frac{.25}{4}\times 3=.1875$$ Decimals of an inch. Lift of the required valve.

Performing the operation, we have:

```
4.00).2500 (0.0625
      2400       3
      ----   -----
      1000   .1875  Decimals of an inch. Lift of required valve.
       800
      ----
      2000
      2000
      ----
```

ANOTHER METHOD.

RULE.—Multiply the lift of the given valve by its diameter, and divide the product by the diameter of the required valve and the quotient will give the lift of the required valve.

Example.—Let 25 one hundredths of an inch equal lift of the given valve.
Let 3 inches equal diameter of the given valve.
Let 4 inches equal diameter of the required valve.

Then we have:

$$\frac{.25 \times 3}{4} = .1875$$ Decimals of an inch. Lift of the required valve.

Performing the operation we have:

```
        .25
          3
       ----
4.00)  .750 (.1875  Decimals of an inch. Lift of the required valve.
        400
       ----
       3500
       3200
       ----
        3000
        2800
        ----
         2000
         2000
         ----
```

LIFT OF FLAT-SEATED VALVES REQUIRED TO PRODUCE A GIVEN AREA OF OPENING.

TO DETERMINE THE LIFT OF A REQUIRED FLAT-SEATED VALVE THAT WILL PRODUCE AN AREA OF OPENING EQUAL TO THE AREA OF THE DIAMETER OF A GIVEN VALVE OR AREA OF CROSS SECTION OF A GIVEN PIPE.

RULE.—Divide the area of the given valve, or area of cross section of the given pipe, by the diameter of the required valve and divide the quotient by 3.1416, *and the last quotient will give the required lift of the required valve to produce an area of opening equal to the area of the diameter of a given valve, or the area of the diameter of a given pipe.*

Example.—What must be the lift of a required valve 5 inches in diameter to produce an area of opening equal to the area of a 3 inch valve or 3 inch pipe?

Let 3 inches equal diameter of a given valve or pipe.
Let .7854 equal a constant.
Let 5 inches equal diameter of the required valve.
Let 3.1416 equal a constant.

Then we have:

$$\left(\frac{3\times3\times.7854}{5}\right)\div3.1416=45 \text{ Hundredths of an inch. Lift of required valve.}$$

Or:

$$\frac{3\times3\times.7854}{5\times3.1416}=45 \text{ Hundredths of an inch. Lift of required valve.}$$

Performing the operation, we have:

```
                3
                3
                -
                9
            .7854
           ------
        5) 7.0686
         --------
3.14160) 1.413720 (0.45  Hundredths of an inch. Lift of required valve.
         1 256640
         --------
           1570800
           1570800
         ---------
```

It will be observed that the area of the diameter of a 3 inch valve or a diameter of a 3 inch pipe is 7.0686 square inches. Then, to determine whether or not a 5 inch valve, lifting 45 one hundredths of an inch, will produce an area of opening equal to the area of the diameter of a 3 inch valve or pipe, let the area of opening produced by a 5 inch valve, lifting 45 one hundredths of an inch, be determined.

RULE.—Multiply the constant 3.1416 *by the diameter of the valve and then multiply the product by the lift of the valve, and the last product will give the area of opening produced by the lift of the valve.*

Example.—Let 5 inches equal diameter of the valve.
Let 3.1416 equal a constant.
Let 45 one hundredths of an inch equal lift of valve.

Then we have:

```
                          3.1416
                               5
                         -------
Am't carried forward,    15.7080
```

Am't brought forward, 15.7080	
.45	
785400	
6 28320	
7.068600	Square inches, area of opening produced by the lift of the valve, and this is exactly the area of the diameter of a 3 inch valve or pipe.

ANOTHER METHOD.

RULE.—First, square the diameter of the pipe and multiply the product by .7854, the last product will give the area of cross section of the pipe, and we set that aside and call it "Product No. 1."

Second, multiply the diameter of the given valve by 3.1416, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2" and the quotient will give the required lift of a given valve to produce an area of opening equal to the area of the diameter of a given pipe.

Example.—Let 8 inches equal diameter of given pipe.
Let .7854 equal a constant.
Let 9 inches equal diameter of given valve.
Let 3.1416 equal a constant.

Then we have:

$$\frac{8\times 8\times .7854}{9\times 3.1416}=1.77+ \text{ inches.}$$ A fraction over 1¾ inches; required lift of the given valve.

Performing the operation in the ordinary way, we have:

8	Diameter of the pipe.
8	Diameter of the pipe.
64	Square of diameter of the pipe.
.7854	A constant.
3 1416	
47 124	
50.2656	"Product No. 1."

Next, multiplying the diameter of the given valve by 3.1416, we have: $9\times 3.1416=28.2744$, "Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

28.2744) 50.2656 (1.77+ inches.	Required lift of given valve.
28 2744	
21 99120	
19 79208	
2 199120	
1 979208	

DIAMETER OF FLAT-SEATED VALVES.

TO DETERMINE THE REQUIRED DIAMETER OF A REQUIRED FLAT-SEATED VALVE TO HAVE A GIVEN LIFT THAT WILL PRODUCE AN AREA OF OPENING EQUAL TO THAT PRODUCED BY THE GIVEN LIFT OF A GIVEN VALVE.

RULE.—Multiply the diameter of the given valve by its lift, and divide the product by the required lift of the required valve.

Example.—Let 2 inches equal diameter of the given valve.
Let 15 one hundredths of an inch equal lift of given valve.
Let 6 one hundredths of an inch equal required lift of required valve.

Then we have:

$$\frac{2\times.15}{.06}=5 \text{ inches. Required diameter of required valve.}$$

Performing the operation, we have:

$$\begin{array}{r} .15 \\ 2 \\ \hline .06)\ .30\ (5 \\ 30 \\ \hline \end{array}$$

inches. Required diameter of required valve.

LIFT OF BEVEL-SEATED VALVES.

The following rules are based upon the United States Standard valves; which valves have their seats made at an angle of 45 degrees to the center line of the valve's axis.

TO DETERMINE THE LIFT OF A BEVEL-SEATED VALVE WHEN THE AREA OF OPENING REQUIRED AND THE DIAMETER OF THE VALVE ARE GIVEN.

RULE.—First, multiply 3.1416 *by the diameter of the valve and call the product "Product No. 1."*

Second, divide the given area of opening required by "Product No. 1," and divide the quotient by .707, *the last quotient will give the required lift of the valve.*

Example.—Let 4.10 inches equal diameter of the valve.
Let 3.1416 equal a constant.
Let 1.821311184 square inches equal area of opening required.
Let .707 equal a constant.

Then we have:

$$\left(\frac{1.821311184}{3.1416\times 4.10}\right)\div .707 = 20 \text{ Hundredths of an inch. Lift required.}$$

Performing the operation in the ordinary way, we have:

```
   3.1416   A constant.
     4.10   Diameter of valve.
  -------
  3141 60
12 5664
---------
12.8805 60  "Product No. 1."
```

Next, dividing the area of opening required by "Product No. 1," we have:

```
12.88056) 1.821311184 (.1414
          1 288056
          --------
           5332551
           5152224
           -------
            1803277
            1288056
            -------
             5152224
             5152224
             -------
```

Finally, dividing the quotient .1414 by .707, we have:

```
.707) .1414 (.2   Tenths, or 20 hundredths of an inch, lift
       1414            required for a valve 4 inches in diame-
       ----            ter to produce an area opening of
                       1.821311184 square inches.
```

DIAMETER OF BEVEL-SEATED VALVES.

TO DETERMINE THE DIAMETER OF A REQUIRED BEVEL-SEATED VALVE TO HAVE A GIVEN LIFT THAT WILL PRODUCE AN AREA OF OPENING EQUAL TO THAT PRODUCED BY THE GIVEN LIFT OF A GIVEN VALVE.

RULE.—First, multiply the given lift of the required valve by .707 *and call the product* "*Product No.* 1."

Second, divide the area of opening produced by the lift of the given valve by "*Product No.* 1" *and call the quotient* "*Quotient No.* 1."

Third, divide "*Quotient No.* 1" *by* 3.1416, *and deduct one-half of the lift of the required valve from the last quotient and the remainder will give the diameter of the required valve*

Example.—Let .25 equal lift of required valve, in hundredths of an inch.

Let 4 equal diameter of given valve, in inches.

Let .20 equal lift of given valve, in hundredths of an inch.

Let 3.1416 equal a constant.

Let .707 equal a constant.

Then we have:

$\overline{4+(.20\div 2)}\times 3.1416\times .20\times .707=1.821311184$ square inches. Area of opening produced by the lift of the given valve.

We will now proceed according to the rule.

First, multiply the given lift of the required valve by .707, thus: $.25\times .707=.17675$, "Product No. 1."

Second, divide the area of opening, 1.821311184, produced by the lift of the given valve, by "Product No. 1." Thus: $1.821311184\div .17675 =10.304448$, "Quotient No. 1."

Third, divide "Quotient No. 1" by 3.1416 and deduct one-half of the lift of the required valve from the last quotient. Thus: $(10.304448 \div 3.1416)-(.25\div 2)=3.155$ inches, diameter of required valve.

CHAPTER IV.

BOILER PLATE.

All boiler plate employed in the construction of steam boilers should possess two qualities which are absolutely necessary to render the use of boilers safe. The first is tensile strength sufficient to withstand the strain put upon it by the pressure of steam; and the second is ductility to prevent cracking by unequal expansion and contraction, and by repeated heating and cooling. The latter quality is just as essential to safety in the use of steam boilers as the former; and therefore no plate of iron or steel should be put into any steam boiler that does not possess the requisite strength and ductility. To the absence of one of these essential qualities, and in many cases both, a large number of disastrous steam boiler explosions are directly attributable, resulting in terrible sacrifices of human life and destruction of enormous amounts of property. An investigation into the cause of such disasters usually follows, and the very convenient conclusion is almost invariably reached that, "low water was the cause of the accident."

Hence, the responsibility is fastened upon the engineer who had charge of the boilers, when in fact the blame should be put upon the boiler maker who built the boilers, or upon the inspector who inspected the material, for allowing such material to enter the construction of the boilers.

It is very essential, therefore, that the boiler maker, as well as the engineer, should thoroughly understand the qualities necessary to render boiler plate fit for use in steam boilers, in order to guard against explosions from the employment of unfit and dangerous material. In order, then, to prevent the use of such material every plate should be tested before putting it into the boiler. The United States Government has fully recognized the importance of guarding against the use of bad material in the construction of marine boilers, and laws have been enacted to regulate the tensile strength and ductility of boiler material and to provide for testing every plate, subjected to tensile strain, employed in the construction of marine boilers. These laws also provide that boiler manufacturers shall, for marine purposes, use material containing a certain percentage of ductility for all boiler plate, based upon the tensile strength per square inch of the plate. And all

plates made for marine purposes are required to be stamped by the manufacturer, at their diagonal corners, about 4 inches from the edges, and at or near the center of the plate, with the tensile strength of the material, the name of the manufacturer, and the place where manufactured, and to have a coupon attached, to be cut off and used for testing the material in each plate, as shown in Fig. 58.

SHEET WITH COUPON ATTACHED.

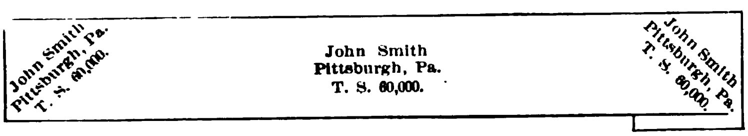


Fig. 58

The coupon is cut off for the use of inspectors in testing the material, and if the test made with the coupon, shows that the material in the plate possesses the required ductility, and the tensile strength stamped on the plate, the plate from which the coupon was cut is approved for use in a marine boiler; and the steam pressure per square inch allowed in such boiler is based upon the tensile strength of the material, the diameter of the boiler, and the manner in which the boiler is constructed. Preparatory to making the test with the coupon, the coupon is cut from the plate and cut out at the center to a width corresponding with the thickness of the material in the coupon. Test pieces $\frac{5}{16}$ inch and under, should be reduced, at point intended for breaking, to one inch in width. Test pieces over $\frac{5}{16}$ inch thick, should be reduced in width at the center so as to approximate an area of cross section at point intended for breaking of $\frac{4}{10}$ of one square inch; but such reduced section should not be less than $\frac{35}{100}$ inch in any case.

FORMS AND DIMENSIONS OF COUPONS.

The following diagram (Fig. 59) shows the form and dimensions of a coupon employed in making a test of the material in the plate from which it was taken.

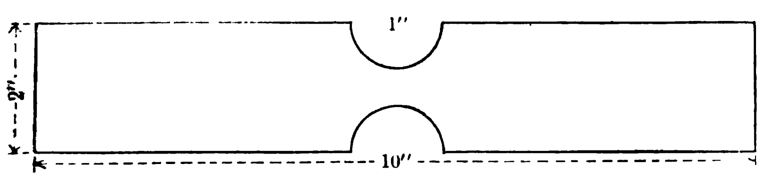


Fig. 59

After the coupon has been cut to the proper width, the first thing to be done is to carefully measure the width and thickness at the smallest part, the part intended for breaking, and make a record of the dimensions.

The next step in the test is to place the coupon in the testing machine, and pull the coupon apart, as shown in the following diagram (Fig. 60):

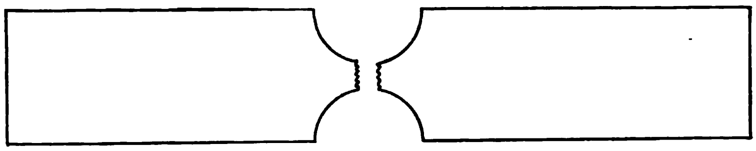


Fig. 60

Dimensions of coupon at point of fracture before breaking.

Width equal .50 of an inch; thickness equal .50 of an inch.

Dimensions of coupon at point of fracture after breaking.

Width equal .36 of an inch; thickness equal .34 of an inch.

Strain at which the coupon parted at point of fracture, 15,000 pounds.

TO DETERMINE THE TENSILE STRENGTH, PER SQUARE INCH, OF THE MATERIAL IN THE COUPON.

RULE.—First, multiply the width of the sample, in hundredths of an inch, at point of fracture before breaking, by the thickness, in hundredths of an inch, at that point, and the product will give the area of the cross section of material at point intended for fracture, in hundredths of an inch.

Second, divide the number of pounds of strain at which the coupon parted, by the area of the material before breaking, and the quotient will give the tensile strength, per square inch, of the material in pounds.

Example.—Let 50 one hundredths of an inch equal width of material before breaking.

Let 50 one hundredths of an inch equal thickness of material before breaking.

Let 15,000 pounds equal strain at which the material parted.

Then we have: $15000 \div (.50 \times .50) = 60000$ lbs. Tensile strength of material per square inch.

Performing the operation in the ordinary way, we have:

.50	Width of material before breaking.
.50	Thickness of material before breaking.
.2500	Area of sample at point of fracture before breaking.

And the strain at which the sample parted, divided by the area of sample at point of fracture, before breaking, we have·

.2500) 15000.0000 (60000 lbs. Tensile strength.
15000
0000

It will be observed that four ciphers have been annexed to the dividend, to correspond with the number of decimals in the divisor, the reasons for which have been explained in a preceding chapter.

TO DETERMINE DUCTILITY OF BOILER PLATE.

RULE.—First, multiply the width of the sample before breaking, in hundredths of an inch, by the thickness, in hundredths of an inch, and the product will give the area of the material at point of fracture before breaking.

Second, multiply the width of the sample at point of fracture after breaking, in hundredths of an inch, by the thickness at point of fracture, in hundredths of an inch, and the product will give the area of material at point of fracture after breaking.

Third, subtract the area of material at point of fracture after breaking, from the area of material at point of fracture before breaking, and call the product "The Remainder."

Fourth, divide "The Remainder" by the area of material at point of fracture before breaking, and the quotient will give the percentage of ductility.

Example.—Let .50 equal width of sample, in hundredths of an inch, before breaking.
Let .50 equal thickness of sample, in hundredths of an inch, before breaking.
Let .36 equal width of sample, in hundredths of an inch, after breaking.
Let .34 equal thickness of sample, in hundredths of an inch, after breaking.

Then we have:

$$\frac{(.50\times.50)-(.36\times.34)}{.50\times.50}=.51+ \text{ Per cent. ductility.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl} .50 & \text{Width of sample before breaking.} \\ .50 & \text{Thickness of sample before breaking.} \\ \hline .2500 & \text{Area of sample before breaking.} \\ \\ .36 & \text{Width of sample after breaking.} \\ .34 & \text{Thickness of sample after breaking.} \\ \hline 144 & \\ 108 & \\ \hline .1224 & \text{Area of sample after breaking.} \end{array}$$

And, deducting the area of the sample at point of fracture after breaking from the area of the sample before breaking, we have:

$$\begin{array}{rl} .2500 & \\ .1224 & \\ \hline .1276 & \text{"The Remainder."} \end{array}$$

Then, dividing "The Remainder" by the area of sample at point of fraction before breaking, we have:

```
.2500) .12760 (0.51+ Per cent. ductility.
       12500
       -----
         2600
         2500
         ----
```

Giving the rule in more condensed form:

RULE.—Subtract the area of sample at point of fracture, after breaking, from the area of sample intended for fracture, before breaking, and then divide "The Remainder" by the area of sample before breaking, and the quotient will give the percentage of ductility of the material.

To make it still plainer:

RULE.—Subtract the reduced area of sample from the original area, and divide "The Remainder" by the original area, the quotient will give the percentage of ductility.

Example.—Let .96 equal original width of sample.
Let .26 equal original thickness of sample.
Let .81 equal reduced width of sample.
Let .15 equal reduced thickness of sample.

Then we have: $.96 \times .26 = .2496$, Original area.
$.81 \times .15 = .1215$, Reduced area.

```
.2496) .12810 (0.513+ Per cent. ductility.
       12480
       -----
         3300
         2496
         ----
          8040
          7488
          ----
```

THE PERCENTAGE OF DUCTILITY REQUIRED IN BOILER PLATE.

No iron plates should be allowed to enter into the construction of any steam boiler, with a ductility of less than 25 per cent., and no steel plates should be allowed in the construction of any steam boiler, with a ductility of less than 50 per cent. for material having a tensile strength of 60,000 pounds, and one per cent. additional for every 1000 pounds additional tensile strength of the material. It will therefore be seen that all iron plates should have a ductility of not less than 25 per cent., and that steel plates should have a ductility of not less than 50 per cent., for 60,000 tensile strength and under, and one per cent. additional for each 1000 pounds in the tensile strength of the steel plates over 60,000 pounds tensile strength. And no steel plate should be used in any steam boiler with a tensile strength of less than 50,000 pounds per square inch.

CHAPTER V.

STEAM BOILERS.

SAFE-WORKING PRESSURE—SINGLE-RIVETED BOILERS.

TO DETERMINE THE SAFE-WORKING PRESSURE FOR A CYLINDRICAL BOILER WITH SINGLE-RIVETED LONGITUDINAL SEAMS.

RULE.—Multiply the thickness of material in the weakest plate in the shell of the boiler, in hundredths of an inch, by the tensile strength, in pounds per square inch, of the material in that plate, then divide the product by one-half of the diameter of the boiler in inches, and divide the quotient by 6, the factor of safety, and the last quotient will give the safe-working pressure, per square inch for a boiler with longitudinal seams single-riveted.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of plates.
Let 40 inches equal diameter of the boiler.
Let 6 equal a factor of safety.

Then we have:

$$\frac{(.25\times 60000)\div(40\div 2)}{6}=125 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have;

$$\begin{array}{r l}
 & \;.25 \\
 & \;\underline{60000} \\
40\div 2= 20) & \;15000.00\;(750 \\
 & \;\underline{140} \\
 & \;\;100 \\
 & \;\;\underline{100} \\
 & \;\;\;\;0
\end{array}$$

And, $750\div 6=125$ lbs. Safe-working pressure.

BURSTING PRESSURE OF SINGLE-RIVETED BOILERS.

TO DETERMINE THE BURSTING PRESSURE OF A BOILER WITH SINGLE-RIVETED LONGITUDINAL SEAMS.

RULE.—Multiply the thickness of the weakest plate in the shell of the boiler, in hundredths of an inch, by the tensile strength in pounds per square

inch, and divide the product by one-half of the diameter of the boiler, in inches, then multiply the quotient by .56, the last product will give the bursting pressure per square inch.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of plates.

Let 40 inches equal diameter of the boiler.

Let .56 equal a constant.

Then we have:

$$\frac{.25\times 60000}{40\div 2}\times .56 = 420 \text{ lbs.}$$ Bursting pressure, per square inch, for single-riveted boilers.

Performing the operation in the ordinary way, we have:

```
              .25
             60000
             ---------
40÷2=  20) 15000.00( 750
            140        .56
            ----     ------
            100       45 00
            100      375 0
            ----     ------
                     420.00 lbs.
```

Bursting pressure, per square inch, for single-riveted boilers.

THICKNESS OF PLATE FOR SINGLE-RIVETED BOILERS.

TO DETERMINE THE THICKNESS OF MATERIAL REQUIRED FOR SAFE-WORKING PRESSURE OF A BOILER, WITH SINGLE-RIVETED LONGITUDINAL SEAMS, WHEN THE DIAMETER, TENSILE STRENGTH OF MATERIAL AND WORKING PRESSURE ARE GIVEN.

RULE.—Multiply the given pressure, per square inch, by 6, and multiply the product by one-half of the diameter of the boiler, and divide the last product by the tensile strength of the material in pounds per square inch, the quotient will give the required thickness of the material in hundredths of an inch.

Example.—Let 125 pounds per square inch equal the given pressure.

Let 6 equal a constant.

Let 40 inches equal diameter of the boiler.

Let 60,000 pounds per square inch equal tensile strength of material.

Then we have:

$$\frac{125\times 6\times (40\div 2)}{60000} = .25$$ Thickness of material required.

Performing the operation in the ordinary way, we have:

	125	Given pressure per square inch.
	6	A constant.
	750	
40÷2=	20	One-half of the diameter of the boiler.

Tensile strength of material. 60000) 150000 (0.25 Thickness of material required.
120000
300000
300000

DIAMETER OF SINGLE-RIVETED BOILERS.

TO DETERMINE THE REQUIRED DIAMETER OF A BOILER, WITH SINGLE-RIVETED LONGITUDINAL SEAMS, WHEN THE WORKING-PRESSURE, THICKNESS AND TENSILE STRENGTH OF MATERIAL ARE GIVEN.

RULE.—First, multiply the given thickness of the material, in hundredths of an inch, by the given tensile strength, per square inch, and call the product "Product No. 1."

Second, multiply the given pressure, per square inch, by 6, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and multiply the quotient by 2, the last product will give the required diameter of the boiler in inches.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of material.

Let 125 pounds equal given pressure per square inch.

Let 6 equal a constant.

Then we have:

$$\frac{.25\times 60000}{125\times 6}\times 2=40 \text{ inches.}$$ Required diameter of the boiler.

Performing the operation, we have:

.25	Thickness of plate.
600 00	Tensile strength of plate.
15000.00	"Product No. 1."

Next we have:

125	Pressure per square inch.
6	Factor of safety.
750	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," and multiplying the quotient by 2, we have:

$$\begin{array}{r} 750)\ 15000\ (20 \\ 1500\quad\ \ 2 \\ \hline 0\quad 40 \end{array}$$

inches. Required diameter of the boiler.

TENSILE STRENGTH OF MATERIAL FOR SINGLE-RIVETED BOILERS.

TO DETERMINE THE REQUIRED TENSILE STRENGTH OF MATERIAL FOR A BOILER, WITH SINGLE-RIVETED LONGITUDINAL SEAMS, WHEN THE DIAMETER, THICKNESS OF MATERIAL, AND PRESSURE PER SQUARE INCH ARE GIVEN.

RULE.—First, multiply the given pressure, per square inch, by 6, and call the product "Product No. 1."

Second, multiply "Product No. 1" by one-half of the given diameter of the boiler, and call the product "Product No. 2."

Third, divide "Product No. 2" by the given thickness of the material, in hundredths of an inch, and the quotient will give the required tensile strength of the material in pounds per square inch of section.

Example.—Let 125 pounds per square inch equal given pressure.
Let 6 equal a constant.
Let 40 inches equal given diameter of the boiler.
Let 25 one hundredths of an inch equal given thickness of material.

Then we have:

$$\frac{125 \times 6 \times (40 \div 2)}{.25} = 60000 \text{ lbs.}$$

Required tensile strength per sectional square inch of material.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 125 \\ 6 \\ \hline 750 \\ 20 \\ \hline .25)\ 15000.00\ (60000 \text{ lbs.} \\ 150\qquad\qquad\qquad\quad \\ \hline 00\,00\qquad\qquad\qquad\quad \end{array}$$

Required tensile strength per sectional square inch of material.

SAFE-WORKING PRESSURE OF DOUBLE-RIVETED BOILERS.

TO DETERMINE THE SAFE-WORKING PRESSURE FOR A CYLINDRICAL BOILER, WITH DOUBLE-RIVETED LONGITUDINAL SEAMS, AND ALL HOLES IN THE PLATES DRILLED INSTEAD OF PUNCHED.

RULE.—Multiply the thickness of the material in the weakest plate in the shell of the boiler, in hundredths of an inch, by the tensile strength of the

material, in pounds per square inch, then divide the product by one-half of the diameter of the boiler, in inches; then divide the quotient by 6, and add 20 per cent. to the last quotient, and the sum will give the safe-working pressure for a cylindrical boiler, with longitudinal seams, double-riveted, and all holes in the plates drilled.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of the plates.
Let 40 inches equal diameter of the boiler.
Let 6 equal a factor of safety.
Let 20 per cent. equal amount to be added for double riveting.

Then we have:

$$\frac{(.25\times 60000)\div(40\div 2)}{6}\times 1.2=150 \text{ lbs.}$$ Pressure per square inch allowable as safe-working pressure.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r}
.25 \\
60000 \\
\hline
40\div 2= \quad 20)\,15000.00 \\
\hline
6)\,750 \\
\hline
125 \\
1.2 \\
\hline
250 \\
125 \\
\hline
150.0 \text{ lbs.}
\end{array}$$

Per cent. annexed to unity and multiplying, is the same as adding the percentage.

Pressure allowable.

SAFE-WORKING PRESSURE—A SIMPLE RULE.

A simple rule for determining the safe-working pressure for boilers with double-riveted longitudinal seams, and one which dispenses with the trouble of adding the 20 per cent. to the pressure, and yet produces exactly the same result in every case, is as follows:

RULE.—Multiply the thickness of material, in hundredths of an inch, in the weakest plate in the shell of the boiler, by the tensile strength of the material, per square inch; then divide the product by one-half of the diameter of the boiler, in inches, and then divide the quotient by 5, the last quotient will give the safe-working pressure, per square inch, in pounds

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of the material.
Let 40 inches equal diameter of the boiler.
Let 5 equal a factor of safety.

Then we have:

$$\frac{(.25\times 60000)\div(40\div 2)}{5}=150 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} .25 \\ 60000 \\ \hline 40\div 2 = 20)\,15000 \\ \hline 5)\,750 \\ \hline 150 \text{ lbs. Safe-working pressure.} \end{array}$$

SAFE-WORKING PRESSURE—ANOTHER RULE.

RULE.—Multiply the thickness of material, in hundredths of an inch, in the weakest plate in the shell of the boiler, by the tensile strength, per square inch, of the material, and divide the product by one-half of the diameter of the boiler, in inches, and multiply the quotient by .2 (two-tenths).

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of the material.
Let 40 inches equal diameter of the boiler.
Let 2 tenths equal a constant.

Then we have:

$$\frac{.25\times 60000}{40\div 2}\times .2=150 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rll} & .25 & \\ & 60000 & \\ \hline \text{Half diameter of boiler.}\quad 20) & 15000.00\ (750 & \\ & 140 & .2 \\ \hline & 100 & 150.0 \text{ lbs. Safe-working pressure.} \\ & 100 & \\ \hline & 0 & \end{array}$$

SAFE-WORKING PRESSURE—A SIMPLER RULE.

Perhaps the simplest rule for determining the safe-working pressure for a boiler with double-riveted longitudinal seams is the following:

RULE.—Multiply the thickness of material, in hundredths of an inch, in the weakest sheet in the boiler, by the tensile strength, and then divide the product by the diameter of the boiler in inches, and then multiply the quotient by .4, the last product will give the safe-working pressure per square inch.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of the material.
Let 40 inches equal diameter of the boiler.
Let .4 equal a constant.

Then we have:

$$\frac{.25 \times 60000}{40} \times .4 = 150 \text{ lbs.}$$ Safe-working pressure, per square inch, for double riveting.

Performing the operation in the ordinary way, we have:

.25 Thickness of material.
60000 Tensile strength of material.
Dividing by the diameter of the boiler. 40) 1500.00 (375
120 .4 A constant.
300 150.0 lbs. Safe-working pressure per square inch for double riveting.
280
200
200

All of these rules give exactly the same result. The first rule for double-riveted boilers is the same as the rule for single-riveted boilers, except that 20 per cent. is added to the pressure allowed for single riveting when the longitudinal seams are double riveted. Therefore, whether we add 20 per cent. according to the first rule for double-riveted longitudinal seams, or divide by 5 and omit adding the 20 per cent. according to the second rule, or multiplying the quotient by .2 according to the third rule, or dividing with the diameter of the boiler instead of one-half of the diameter, and multiplying the quotient by .4 according to the fourth rule, it is the same in each case as the adding of 20 per cent. according to the first rule; hence, the student can select either of the foregoing rules with perfect safety, and with the assurance that each is absolutely correct.

BURSTING PRESSURE OF DOUBLE-RIVETED BOILERS.

TO DETERMINE THE BURSTING PRESSURE OF A BOILER WITH DOUBLE-RIVETED LONGITUDINAL SEAMS, AND ALL HOLES IN PLATE DRILLED.

RULE.—Multiply the thickness of the material in the weakest plate in the boiler, in hundredths of an inch, by the tensile strength in pounds per square inch, and divide the product by one-half of the diameter of boiler in inches, and multiply the quotient by .70, the last product will give the bursting pressure in pounds per square inch.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of material.

Let 40 inches equal diameter of the boiler.

Let .70 equal a constant.

Then we have:

$$\frac{.25\times 60000}{40\div 2}\times .70=525 \text{ lbs.}$$ Pressure per square inch required to burst the boiler.

Performing the operation in the ordinary way, we have:

```
               .25
              60000
             ———————
40÷2=  20) 15000.00 (750
           140          .70
           ———         ————
            100       525.00 lbs. Bursting pressure.
            100
            ———
```

SIMPLE RULE FOR DETERMINING BURSTING PRESSURE.

A simpler rule for determining the bursting pressure, and one that gives exactly the same result, is as follows:

RULE.—Multiply the thickness of material, in hundredths of an inch, of the weakest plate in the shell of the boiler by the tensile strength of the material, per square inch, and then divide the product by the diameter of the boiler, in inches, and then multiply the quotient by 1.4, the last product will give the bursting pressure, in pounds per square inch, for double-riveted longitudinal seams.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of material.

Let 40 inches equal diameter of the boiler.

Let 1.4 equal a constant.

Then we have:

$$\frac{.25\times 60000}{40}\times 1.4=525 \text{ lbs.}$$ **Bursting Pressure.**

Performing the operation in the ordinary way, we have:

```
        .25
      60000
    -------
40) 15000.00 (375
    120         1.4
    ---        ---
     300       150 0
     280       375
     ---       -----
      200      525.0 lbs.   Bursting pressure per square inch.
      200
      ---
```

THICKNESS OF PLATE FOR DOUBLE-RIVETED BOILERS.

TO DETERMINE THE THICKNESS OF MATERIAL REQUIRED FOR SAFE-WORKING PRESSURE OF A BOILER WITH DOUBLE-RIVETED LONGITUDINAL SEAMS, WHEN THE DIAMETER, TENSILE STRENGTH OF MATERIAL AND WORKING PRESSURE ARE GIVEN.

RULE.—Multiply the given pressure, per square inch, by 5, then multiply the product by one-half of the given diameter of the boiler, in inches, and then divide the last product by the given tensile strength of the material, and the quotient will give the thickness of the required material in hundredths of an inch.

Example.—Let 150 pounds per square inch equal the given pressure.
Let 5 equal a constant for double-riveted longitudinal seams.
Let 40 inches equal given diameter of the boiler.
Let 60,000 pounds per square inch equal tensile strength of the material.

Then we have:

$$\frac{150\times 5\times (40\div 2)}{60000}=.25$$ **Thickness of required material in hundredths of an inch.**

Performing the operation in the ordinary way, we have:

```
                                         150
                                           5
                                         ---
                                         750
                                40÷2=     20
                                      ------
Tensile strength of material.  60000) 150000 (0.25  Thickness of required material.
                                      120000
                                      ------
                                      300000
                                      300000
                                      ------
```

DIAMETER OF DOUBLE-RIVETED BOILERS.

TO DETERMINE THE REQUIRED DIAMETER OF A BOILER, WITH DOUBLE-RIVETED LONGITUDINAL SEAMS, WHEN THE WORKING PRESSURE, THICKNESS AND TENSILE STRENGTH OF MATERIAL ARE GIVEN.

RULE.—First, multiply the given thickness of material, in hundredths of an inch, by the given tensile strength in pounds per square inch, and call the product "Product No. 1."

Second, multiply the given pressure, per square inch, by 5, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and multiply the quotient by 2, the last product will give the required diameter of the boiler.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of material.

Let 150 pounds per square inch equal given pressure.

Let 5 equal a constant.

Then we have:

$$\frac{.25\times 60000}{150\times 5}\times 2=40 \text{ inches. Required diameter of the boiler.}$$

Performing the operation, we have:

$$\begin{array}{r} .25 \\ 60000 \\ \hline 15000.00 \end{array} \text{ "Product No. 1."}$$

Next we have:

$$\begin{array}{r} 150 \\ 5 \\ \hline 750 \end{array} \text{ "Product No. 2."}$$

Finally we have:

$$\begin{array}{r} 750)\ 15000\ (20 \\ 1500\quad\ 2 \\ \hline 0\quad 40 \end{array} \text{ inches. Required diameter of the boiler.}$$

TENSILE STRENGTH OF MATERIAL FOR DOUBLE-RIVETED BOILERS.

TO DETERMINE THE REQUIRED TENSILE STRENGTH OF MATERIAL FOR A BOILER WITH DOUBLE-RIVETED LONGITUDINAL SEAMS, WHEN THE DIAMETER, THICKNESS OF MATERIAL, AND PRESSURE PER SQUARE INCH ARE GIVEN.

RULE.—First, multiply the given pressure, per square inch, by 5, and call the product "Product No. 1."

Second, multiply "Product No. 1" by one-half of the given diameter of the boiler, and call the product "Product No. 2."

Third, divide "Product No. 2" by the given thickness of the material, in hundredths of an inch, and the quotient will give the required tensile strength, per square inch, of the material in pounds.

Example.—Let 150 pounds per square inch equal the given pressure.
Let 5 equal a constant.
Let 40 inches equal given diameter of the boiler.
Let 25 one hundredths of an inch equal given thickness of material required.

Then we have:

$$\frac{150 \times 5 \times (40 \div 2)}{.25} = 60000 \text{ lbs.}$$ Tensile strength per square inch of material required.

Performing the operation in the ordinary way, we have:

$$\begin{array}{rr}
 & 150 \\
 & 5 \\
\hline
 & 750 \\
40 \div 2 = & 20 \\
\hline
.25) & 15000.00 \; (60000 \text{ lbs.} \\
 & 150 \\
\hline
 & 0000
\end{array}$$

Required tensile strength of material per square inch.

STRAIN ON LONGITUDINAL SEAMS.

TO DETERMINE THE STRAIN PRODUCED ON LONGITUDINAL SEAMS, TENDING TO TEAR THEM ASUNDER, BY ANY GIVEN PRESSURE PER SQUARE INCH.

RULE.—Multiply the given pressure, per square inch, in pounds, by one-half of the diameter of the boiler, in inches, and the product will give the strain, in pounds, on each longitudinal inch along the shell of the entire boiler as well as on the longitudinal seams.

Example.—Let 150 pounds per square inch equal given pressure.
Let 48 inches equal diameter of the boiler.

Then we have:

$$\begin{array}{rrl}
 & 150 & \text{Given pressure,} \\
48 \div 2 = & 24 & \text{Half diameter of the boiler} \\
\hline
 & 600 & \\
 & 300 & \\
\hline
 & 3600 \text{ lbs.} & \text{Strain on each inch along the entire length of the boiler.}
\end{array}$$

This is the same as though the boiler was made of hoops an inch wide, the strain on each hoop tending to burst it would be 3600 pounds, if the hoops were 48 inches in diameter.

LONGITUDINAL STRAIN ON BOILERS.

TO DETERMINE THE STRAIN IN A LONGITUDINAL DIRECTION ON EACH INCH IN THE CIRCUMFERENCE OF THE BOILER, IN A SHELL HAVING NO FLUES OR BRACES.

RULE.—First, square the diameter of the boiler, in inches, and multiply the product by .7854, *and then multiply the last product by the given pressure, per square inch, and call the product "Product No.* 1."

Second, multiply 3.1416 *by the diameter of the boiler, in inches, and call the product "Product No.* 2."

Third, divide "Product No. 1" *by "Product No.* 2" *and the quotient will give the strain in a longitudinal direction on each inch in the circle of the boiler.*

Example.—Let 48 inches equal diameter of the boiler.
Let .7854 equal a constant.
Let 150 pounds per square inch equal given pressure.
Let 3.1416 equal a constant.

Then we have: $\frac{48\times48\times.7854\times150}{3.1416\times48}=1800$ lbs. **Strain in a longitudinal direction on each inch in the circumference of the boiler**

Performing the operation in the ordinary way, we have:

```
         48  Diameter of boiler.
         48  Diameter of boiler.
        ---
        384
       192
       ----
       2304
      .7854
      -----
       9216
    11 520
   184 32
  1612 8
  ---------
  1809.5616  Area of diameter of boiler.
        150  Pressure per square inch.
  ----------
  90478 0800
 180956 16
 -----------
 271434.2400  "Product No. 1."
```

Next, multiplying 3.1416 by the diameter of the boiler, we have:

```
     3.1416  A constant.
         48  Diameter of boiler.
     ------
    25 1328
   125 664
   --------
   150.7968  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
150.7968) 271434.2400 ( 1800 lbs.   Strain in a longitudinal
          150796 8                   direction on each
          --------                   inch in the circum-
          120637 44                  ference of the boiler.
          120637 44
          ---------
```

Supposing the boiler to be made of 151 staves, extending the entire length of the boiler, and 150 of them would be each one inch in width, and one would be .7968 of an inch in width—a fraction over $\frac{3}{4}$ of an inch—each of the staves or strips an inch wide would have a strain on them lengthwise of 1800 pounds, and the odd strip would have a strain in proportion to its width.

The rule given is the general rule laid down for the guidance of engineers and boiler makers, and while it is absolutely correct, it is a long, tedious and very cumbersome rule. We will, therefore, give another rule which is also absolutely correct, and while it gives the answer exactly the same as the other rule in every case, it has a decided advantage on account of its brevity.

SIMPLE RULE FOR DETERMINING LONGITUDINAL STRAIN ON BOILERS.

RULE.—Divide the diameter of the boiler by 4, and multiply the quotient by the given pressure, per square inch, and the product will give the total strain in a longitudinal direction on each inch in the circumference of a boiler having no flues or braces.

Example.—Let 4 equal a constant.
Let 48 inches equal diameter of the boiler.
Let 150 pounds per square inch equal given pressure.

Then we have:

```
4)48  Diameter of boiler.
  12
  150  Given pressure per square inch.
  ---
  600
 12
 ----
 1800 lbs.  Strain in a longitudinal direction on
            each inch in the circumference of
            the boiler.
```

WEIGHT OF STEAM DISCHARGED UNDER ANY GIVEN PRESSURE THROUGH A PIPE OR ORIFICE OF A GIVEN DIAMETER.

A knowledge of the rule for determining the weight of steam a given pipe or orifice will discharge into the atmosphere, under any given pressure, in a given time, is very important to engineers, as it

frequently happens that owners of steam boilers furnish their neighbors or tenants, steam for power or heating purposes. It is, therefore, essential that the engineer should be able to determine the amount of coal required to generate the amount of steam required. We will, therefore, lay down a simple rule, which will be sufficiently accurate for all practical purposes.

RULE.—Divide the steam pressure, per square inch, carried in the boiler by 70, *and multiply the area of the orifice or cross section of the pipe, to be employed for conveying the steam, by the quotient, and the product will give the weight of steam discharged in pounds per second.*

Example.—Let 105 pounds equal steam pressure per square inch.
Let 70 equal a constant.
Let 1.5 inch equal diameter of pipe or orifice.
Let .7854 equal a constant.

Then we have:

$1.5^2 \times .7854 \times (105 \div 70) = 2.6507250$ pounds. Weight of steam discharged per second.

Performing the operation, we have:

```
70) 105 (1.5   The quotient.
     70
    ---
     350
     350
    ---
```

Next we have:

```
       1.5
       1.5
      ----
        75
      1 5
      ----
      2.25
     .7854
    ------
     39270
    15708
   1 5708
   --------
   1.767150 square inches.   Area of orifice or cross section of pipe.
```

Finally, multiplying the area of the orifice or cross section of the pipe by the quotient, we have:

```
   1.767150
        1.5
   ---------
    8835750
   1 767150
   ---------
   2.6507250 lbs.   Weight of steam discharged per second.
```

Then, to determine the weight of steam discharged per hour, we multiply the weight discharged per second by 3600, thus:

$$
\begin{array}{r}
2.6507250 \\
3600 \\
\hline
1590\ 4350000 \\
7952\ 1750 \\
\hline
9542.6100000
\end{array}
\text{ lbs.}
$$

Weight of steam discharged per hour into the atmosphere.

This amount of steam does not represent the amount of steam a tenant actually uses, but it does represent the amount he can use if furnished steam through a 1½ inch pipe. The cost of furnishing that amount of steam depends upon the efficiency of the boiler, and whether the feed water is heated before it enters the boiler. If the feed water is heated by exhaust steam, the amount of heat imparted to the water is clear gain. If the feed water is heated by live steam, the amount of heat imparted to it is imparted at the expense of the owner of the boiler.

The ordinary two-flue boiler, with a well-proportioned furnace and stack, will evaporate, ordinarily, from 6 to 7 pounds of water per pound of good coal, with the temperature of feed water at 60° Fahrenheit, and 7 to 8 pounds with feed water at 200° Fahrenheit.

A well-proportioned tubular boiler, with properly proportioned furnace and stack, will evaporate from 8 to 9 pounds of water per pound of good coal, with a feed water temperature of 60° Fahrenheit, and from 9 to 10 pounds with a feed water temperature of 200° Fahrenheit.

CHAPTER VI.

RIVETED JOINTS.

DISTANCE OF RIVET HOLES FROM EDGE OF PLATE.

The center of outer rows of rivet holes, for single or double riveting, should in no case be less than one and one-half times the diameter of the rivet holes from the edge of plate.

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.
Let 1.5 equal a constant.

Then we have:

.75	Diameter of rivet hole.
1.5	A constant.
375	
75	
1.125 inches.	Distance of center of rivet hole from edge of the sheet.

SHEARING STRENGTH OF RIVETS.

The shearing strength of iron rivets of a given area of cross section is very nearly equal to the tensile strength of iron plate of equal area of cross section, and they are so taken in the examples. But the shearing strength of steel rivets employed with steel plates is from 15 to 20 per cent. less than the tensile strength of the plate; therefore, due allowance has been made in the rules and examples relating to steel plates and steel rivets.

DISTANCE BETWEEN CENTERS OF ROWS OF RIVET HOLES FOR CHAIN RIVETING.

TO DETERMINE THE DISTANCE BETWEEN CENTERS OF ROWS OF RIVET HOLES.

RULE.—Multiply the diameter of rivet hole by the constant 2.5, *and the product will give the distance between centers of rows of chain rivet holes.*

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.
Let 2.5 equal a constant.

Then we have: $.75 \times 2.5 = 1.875$, or $1\frac{7}{8}$ inches. Distance between centers of rows of rivet holes for chain riveting.

Performing the operation, we have:

$$
\begin{array}{rl}
.75 & \text{Diameter of rivet hole.} \\
2.5 & \text{A constant.} \\
\hline
375 & \\
1\,50 & \\
\hline
1.875, \text{ or } 1\tfrac{7}{8} \text{ inches.} & \text{Distance between centers of rows of rivet holes for chain riveting.}
\end{array}
$$

DISTANCE BETWEEN CENTERS OF ROWS OF RIVET HOLES FOR ZIG-ZAG RIVETING.

TO DETERMINE THE DISTANCE BETWEEN CENTERS OF ROWS OF RIVET HOLES.

RULE.—First, multiply the pitch from center to center of rivet holes by the constant 11, *and add four times the diameter of rivet hole to the product, and call the sum "Sum No.* 1."

Second, add four times the diameter of rivet hole to the pitch from center to center of rivet holes, and call the sum "Sum No. 2."

Third, multiply "Sum No. 1" *by "Sum No.* 2" *and extract the square root of the product; then divide the square root by the constant* 10, *and the quotient will give the required distance between centers of rows of rivet holes.*

Example.—Let 2.5885 inches equal pitch from center to center of rivet holes.
Let 11 equal a constant.
Let 625 one thousandths of an inch equal diameter of rivet hole.
Let 4 equal a constant.
Let 10 equal a constant.

Then we have:

$$\frac{\sqrt{(11\times 2.5885+4\times .625)\times(2.5885+4\times .625)}}{10}=1.25+ \text{ or } 1\tfrac{1}{4} \text{ inches.}$$

Distance between centers of rows of rivet holes for zig-zag riveting.

Performing the operation, we have:

$$
\begin{array}{lrl}
 & 2.5885 & \text{Pitch of rivet hole.} \\
 & 11 & \text{A constant.} \\
\cline{2-2}
 & 2\,5885 & \\
 & 25\,885 & \\
\cline{2-2}
 & 28\,4735 & \\
\text{Then four times diameter of rivet hole. } .625\times 4= & 2\,500 & \\
\cline{2-2}
 & 30.9735 & \text{"Sum No. 1."}
\end{array}
$$

Next we have:

```
                                            2.5885  Pitch.
Adding four times the       .625 × 4 =      2.500
diameter of rivet hole                     -------
to the pitch.                               5.0885  "Sum No. 2."
```

Next, multiplying "Sum No. 1" by "Sum No. 2" we have:

```
     30.9735
      5.0885
  ----------
     1548675
    2477880
   2 477880
 154 8675
 ------------
 157.60865475  Product.
```

Next, extracting the square root of the product, we have:

```
         157.60865475 ( 12.5542+  Square root.
         1
        ---
    22 ) 57
         44
        ----
   245 ) 1360
         1225
        -----
  2505 ) 13586
         12525
        ------
 25104 ) 106154
         100416
        -------
251082 ) 573875
          502164
```

Finally, dividing the square root by the constant 10, we have:

```
10 ) 12.5542 ( 1.25542   Or a fraction over 1¼ inches, required distance between centers of rows of rivet holes for zig-zag riveting.
     10
     --
      25
      20
      --
       55
       50
       --
        54
        50
        --
         42
         40
         --
          20
          20
          --
```

DIAMETER OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE DIAMETER OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—Extract the square root of the thickness of the plate in decimals of an inch and multiply the square root by the constant 1.25, *and the product will give the required diameter of rivet holes.*

Example.—Let 25 one hundredths of an inch equal thickness of plate.

Let 1.25 equal a constant.

Then we have: $(\sqrt{.25})\times 1.25=.625$, or $\frac{5}{8}$ inch. Diameter of rivet holes.

Performing the operation, we have: $\dot{.25}$ (.5 Square root.
.25

Next, multiplying the square root by the constant 1.25 we have: $.5\times 1.25=.625$, or $\frac{5}{8}$ inch. Diameter of rivet holes.

DIAMETER OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE DIAMETER OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—Extract the square root of the thickness of the plate in decimals of an inch and multiply the square root by the constant 1.125, *and the product will give the required diameter of rivet holes.*

Example.—Let 25 one hundredths of an inch equal thickness of plate.

Let 1.125 equal a constant.

Then we have: $(\sqrt{.25})\times 1.125=.5625$, or $\frac{9}{16}$ inch. Diameter of rivet holes.

Performing the operation, we have: $\dot{.25}$ (.5 Square root.
.25

Next, multiplying the square root by the constant 1.125, we have: $.5\times 1.125=.5625$, or $\frac{9}{16}$ inch. Diameter of rivet holes.

DIAMETER OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE DIAMETER OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Extract the square root of the thickness of the plate in decimals of an inch and multiply the square root by the constant 1.375, *and the product will give the required diameter of rivet holes.*

Example.—Let 25 one hundredths of an inch equal thickness of plate.
Let 1.375 equal a constant.

Then we have: $(\sqrt{.25}) \times 1.375 = .6875$. or $\frac{11}{16}$ inch. Diameter of rivet holes.

Performing the operation, we have: $.\dot{2}5$ (.5 Square root.
.25

Next, multiplying the square root by the constant 1.375, we have: $.5 \times 1.375 = .6875$, or $\frac{11}{16}$ inch. Diameter of rivet holes.

DIAMETER OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE DIAMETER OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Extract the square root of the thickness of the plate and multiply the root by the constant 1.25, *and the product will give the diameter of rivet holes.*

Example.—Let 25 one hundredths of an inch equal thickness of plate.
Let 1.25 equal a constant.

Then we have: $(\sqrt{.25}) \times 1.25 = .625$, or $\frac{5}{8}$ inch. Diameter of rivet holes.

Performing the operation, we have: $.\dot{2}5$ (.5 Square root.
.25

Next, multiplying the square root by the constant 1.25, we have: $.5 \times 1.25 = .625$, or $\frac{5}{8}$ inch. Diameter of rivet holes.

PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

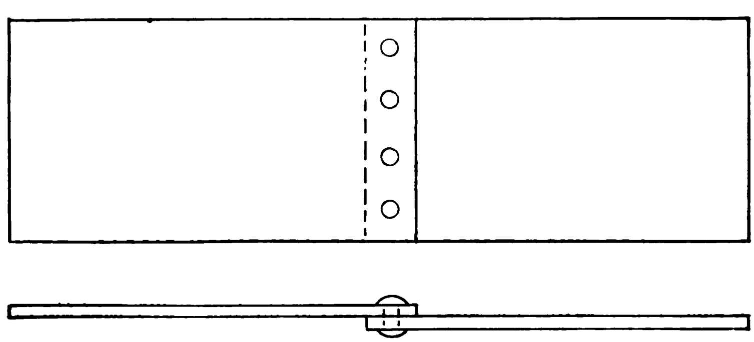


Fig. 61

TO DETERMINE THE PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—Divide the area of diameter of rivet hole by the thickness of the plate in decimals of an inch, then add the diameter of the rivet hole to the quotient, and the sum will give the required pitch from center to center of rivet holes.

Example.—Let $.625 \times .625 \times .7854$ equal area of diameter of rivet hole.

Let 25 one hundredths of an inch equal thickness of plate.

Let 625 one thousandths of an inch equal diameter of rivet hole.

Then we have:

$$\frac{.625 \times .625 \times .7854}{.25} + .625 = 1.852+ \text{ inches.}$$ Pitch of rivet holes from center to center.

Performing the operation in the ordinary way, we have:

```
       .625   Diameter of rivet hole.
       .625   Diameter of rivet hole.
       ----
       3125
      1250
     3750
     ------
    .390625
      .7854
    -------
    1562500
   1953125
  3125000
 2734375
 ----------
 .3067968750  Area of diameter of rivet hole.
```

Then, dividing the area of diameter of rivet hole by the thickness of the plate in decimals of an inch, we have:

```
.250000000).306796875 ( 1.227+   The quotient.
            250000000
            ---------
             567968750
             500000000
             ---------
              679687500
              500000000
              ---------
              179687500
              175000000
              ---------
```

Finally, adding the diameter of rivet hole to the quotient, we have:

1.227	The quotient.
.625	Diameter of rivet hole.
1.852 inches.	Pitch from center to center of rivet hole.

It will be noticed that seven ciphers have been annexed to the thickness of the plate in the divisor; that is done under the rule requiring an equal number of decimals in the divisor and dividend in performing the operation of division with decimal fractions, in order to readily determine the whole numbers in the quotient.

PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—Multiply the area of diameter of rivet hole by the number of rows of rivet holes (2), *then divide the product by the thickness of the plate in decimals of an inch, and add the diameter of rivet hole to the quotient, the sum will give the required pitch of rivet holes in inches.*

Example.—Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let 2 equal number of rows of rivet holes.

Let 3125 ten thousandths of an inch ($\frac{5}{16}$ inch) equal thickness of plate.

Then we have:

$$\frac{.625\times.625\times.7854\times2}{.3125}+.625=2.5885 \text{ inches.}$$ Pitch of rivet holes from center to center

Performing the operation, we have:

.625	Diameter of rivet hole.
.625	Diameter of rivet hole.
3125	
1250	
3750	
.390625	
.7854	
1562500	
1953125	
3125000	
2734375	
Am't carried forward, .3067968750	Area of diameter of rivet hole.

Am't brought forward, .3067968750 Area of diameter of rivet hole.
2 Number of rows of rivet holes.

Next, dividing by the thickness of plate. .3125) .6135937500 (1.9635
3125 .625 Diameter of rivet hole.

30109 2.5885 inches. Required pitch from center to center of rivet holes.
28125

19843
18750

10937
9375

15625
15625

NOTE.—For double-riveted and triple-riveted laps, iron plate and iron rivets, the diameter of rivet holes should be one-sixteenth of an inch less than the single-riveted laps.

PITCH OF RIVET HOLES FOR TRIPLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE PITCH OF RIVET HOLES FOR TRIPLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—First, multiply the area of diameter of rivet hole by the number of rows of rivet holes and call the product "Product No. 1."

Second, divide "Product No. 1" by the thickness of the plate and add the diameter of rivet hole to the quotient, and the sum will give the required pitch of rivets in inches.

Example.—Let 875 one thousandths of an inch ($\frac{7}{8}$ inch) equal diameter of rivet hole.

Let 3 equal number of rows of rivet holes.

Let 75 one hundredths of an inch ($\frac{3}{4}$ inch) equal thickness of plate.

Then we have:

$$\frac{.875\times.875\times.7854\times3}{.75}+.875=3.2802 \text{ inches.}$$ Pitch from center to center of rivet holes.

Performing the operation, we have:

.875 Diameter of rivet hole.
.875 Diameter of rivet hole.

4375
6125
7000

Am't carried forward, .765625

Am't brought forward,

```
  .765625
    .7854
 --------
  3062500
 3828125
6125000
5359375
------------
.6013218750   Area of diameter of rivet hole.
          3   Number of rows of rivet holes.
------------
1.8039656250  "Product No. 1."
```

Dividing "Product No. 1" by the thickness of plate, we have:

```
.75 ) 1.803965625 ( 2.4052
      1 50          .875   Diameter of rivet hole.
      ----         ------
       303         3.2802 inches.  Pitch from center to
       300                         center of rivet
       ---                         holes.
        396
        375
        ---
         215
         150
         ---
```

PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Multiply the area of the diameter of the rivet hole by .8, and divide the product by the thickness of the plate, and add the diameter of the rivet hole to the quotient, and the sum will give the required pitch from center to center of rivet holes.

Example.—Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let .8 equal a constant.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have:

$$\frac{.625 \times .625 \times .7854 \times .8}{.25} + .625 = 1.60675$$ Pitch of rivet holes.

Performing the operation, we have:

```
  .625  Diameter of rivet hole.
  .625  Diameter of rivet hole.
  ----
  3125
 1250
3750
------
.390625
```

Am't carried forward, .390625

Am't brought forward, .390625
.7854

1562500
1953125
3125000
2734375

.3067968750 Area of diameter of rivet hole.
.8 A constant.

.25) .24543750000 (.98175
225 .625 Diameter of rivet hole.

204 1.60675 inches. Pitch of rivet holes from center to center.
200

43
25

187
175

125
125

PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

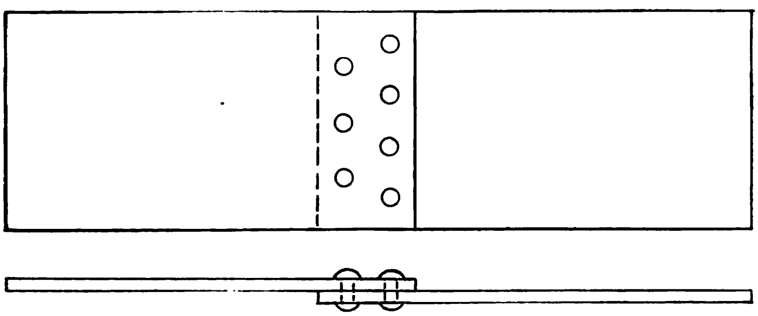


Fig. 62

TO DETERMINE THE PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Multiply the area of diameter of rivet hole by the number of rows of rivet holes (2), *then multiply the product by* .80, *then divide the last product by the thickness of the plate and add the diameter of rivet hole to the quotient, the sum will give the required pitch of rivet holes in inches.*

Example.—Let 625 one thousandths of an inch (⅝ inch) equal diameter of rivet hole.
Let 2 equal number of rows of rivet holes.
Let .80 equal percentage of shearing strength of rivets.
Let 25 one hundredths of an inch equal thickness of plate.

Am't brought forward, .765625

.765625	
.7854	
3062500	
3828125	
6125000	
5359375	
.6013218750	Area of diameter of rivet hole.
3	Number of rows of rivet holes.
1.8039656250	"Product No. 1."

Dividing "Product No. 1" by the thickness of plate, we have:

.75) 1.803965625 (2.4052
1 50   .875 Diameter of rivet hole.
303   3.2802 inches. Pitch from center to center of rivet holes.
300
396
375
215
150

PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE PITCH OF RIVET HOLES FOR SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Multiply the area of the diameter of the rivet hole by .8, and divide the product by the thickness of the plate, and add the diameter of the rivet hole to the quotient, and the sum will give the required pitch from center to center of rivet holes.

Example.—Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let .8 equal a constant.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have:

$$\frac{.625\times.625\times.7854\times.8}{.25}+.625=1.60675$$ Pitch of rivet holes.

Performing the operation, we have:

.625	Diameter of rivet hole.
.625	Diameter of rivet hole.
3125	
1250	
3750	
.390625	

Am't carried forward, .390625

```
Am't brought forward,   .390625
                          .7854
                        -------
                        1562500
                       1953125
                      3125000
                     2734375
                     ------------
                     .3067968750  Area of diameter of rivet hole.
                               .8  A constant.
                     ------------
                .25).24543750000 (.98175
                     225              .625    Diameter of rivet hole.
                     ---              -------
                     204             1.60675 inches.  Pitch of rivet holes
                     200                              from center to center.
                     ---
                      43
                      25
                     ---
                     187
                     175
                     ---
                      125
                      125
                      ---
```

PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

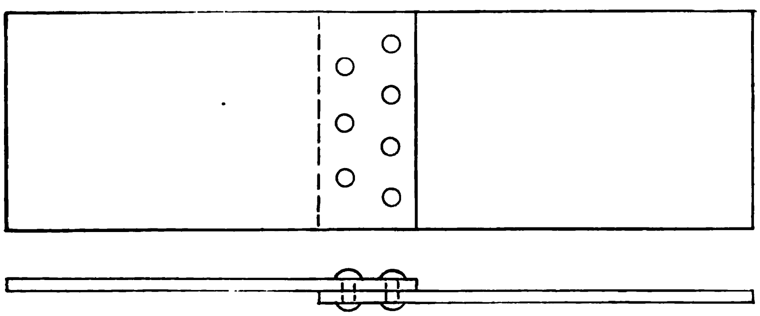


Fig. 62

TO DETERMINE THE PITCH OF RIVET HOLES FOR DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—Multiply the area of diameter of rivet hole by the number of rows of rivet holes (2), *then multiply the product by* .80, *then divide the last product by the thickness of the plate and add the diameter of rivet hole to the quotient, the sum will give the required pitch of rivet holes in inches.*

Example.—Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let 2 equal number of rows of rivet holes.

Let .80 equal percentage of shearing strength of rivets.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have:

$$\frac{.625\times.625\times.7854\times2\times.80}{.25}+.625=2.588+ \text{ inches. Required pitch of rivet holes.}$$

Performing the operation in the ordinary way, we have:

```
             .625  Diameter of rivet hole.
             .625  Diameter of rivet hole.
             ----
             3125
            1250
           3750
           ------
           .390625
             .7854
           -------
           1562500
          1953125
         3125000
        2734375
        ------------
        .3067968750  Area of diameter of rivet hole.
                  2  Number of rows of rivet holes.
        ------------
        .6135937500
                .80  Percentage of shearing strength of rivets.
        ------------
Dividing by thickness of plate. .25).490875000000 (1.963+
           25             .625  Diameter of rivet hole.
           ---           -----
           240           2.588 inches.  Required pitch of rivet holes.
           225
           ---
            158
            150
            ---
              87
              75
              --
```

PITCH OF RIVET HOLES FOR TRIPLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE PITCH OF RIVET HOLES FOR TRIPLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—First, multiply the area of diameter of rivet hole by the number of rows of rivet holes (3), *then multiply the product by the shearing strength per sectional square inch of the rivet, which is* .80 *per cent. of the tensile strength of the plate, and therefore,* .80 *is used as a multiplier, then call the product "Product No.* 1."

Second, divide "Product No. 1" *by the thickness of the plate, and add the diameter of rivet hole to the quotient, the sum will give the pitch, in inches, from center to center of rivet holes in each row.*

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.

Let 3 equal number of rows of rivet holes.

Let .80 equal percentage of shearing strength of rivets as compared with tensile strength of plate.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75\times.75\times.7854\times3\times.80}{.5}+.75=2.87+\text{ inches.}$$ Required pitch of rivet holes.

Performing the operation in the ordinary way, we have:

```
        .75       Diameter of rivet hole.
        .75       Diameter of rivet hole.
      -----
        375
       525
      -----
      .5625
      .7854
      -----
      22500
     28125
    45000
   39375
 -----------
  .44178750       Area of diameter of rivet hole.
          3       Number of rows of rivet holes.
 -----------
 1.32536250
         80       Percentage of shearing strength of rivets
                  as compared with tensile strength
 --------------   of plate.
 1.0602900000     "Product No. 1."
```

Next, dividing "Product No. 1" by the thickness of the plate and adding the diameter of rivet hole to the quotient, we have:

```
.5) 1.06029 (2.12+
    1 0        .75  Diameter of rivet hole.
    ---       -----
      6       2.87 inches.  Required pitch of rivet holes.
      5
     --
     10
     10
     --
```

STRENGTH OF SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF SINGLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch, and divide the remainder by the pitch, and the quotient will give the percentage of strength of plate at joint.

Second, square the diameter of rivet hole and multiply by .7854, *which will give the area of cross section of the rivet, and call the product "Product No. 1."*

Third, multiply the pitch of rivets by the thickness of the plate in decimals of an inch, and call the product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2" and the quotient will give the percentage of strength of rivets at joint."

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 625 one thousandths of an inch equal diameter of rivet hole.

Let 1.852 inches equal pitch from center to center of rivet holes.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have: $$\frac{1.852-.625}{1.852}=.66+$$ Percentage of strength of plate at joint.

Performing the operation, we have:

```
            1.852  Pitch of rivet holes.
             .625  Diameter of rivet hole.
           ------
   1.852 ) 1.2270 ( 0.66+  Percentage of strength of plate at joint.
           1 1112
           ------
            11580
            11112
            -----
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Next we have the percentage of strength of rivets at joint:

$$\frac{.625\times.625\times.7854}{1.852\times.25}=.6626+$$ Percentage of strength of rivets at joint.

Performing the operation, we have:

```
                  .625  Diameter of rivet hole.
                  .625  Diameter of rivet hole.
                 -----
                  3125
                 1250
                3750
                ------
                .390625
                  .7854
               --------
                1562500
               1953125
              3125000
             2734375
             -----------
"Product No. 1."  .3067968750  Area of diameter of rivet hole
```

Next, multiply the pitch of rivets by the thickness of the plate, we have:

$$\begin{array}{rl} 1.852 & \text{Pitch of rivets.} \\ .25 & \text{Thickness of plate.} \\ \hline 9260 & \\ 3704 & \\ \hline .46300 & \text{“Product No. 2.”} \end{array}$$

Finally, dividing "Product No. 1" by "Product No. 2," we have:

$$\begin{array}{rl} .46300).306796875\ (0.6626+ & \text{Percentage of strength of rivets at joint.} \\ 277800 & \\ \hline 289968 & \\ 277800 & \\ \hline 121687 & \\ 92600 & \\ \hline 290875 & \\ 277800 & \\ \hline \end{array}$$

STRENGTH OF DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF DOUBLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—First, subtract the diameter of one rivet hole from the pitch, then divide the remainder by the pitch, and the quotient will give the percentage of strength of the plate at joint.

Second, multiply the area of diameter of one rivet hole by the number of rows of rivets (2), *and call the product "Product No. 1."*

Third, multiply the pitch from center to center of rivets by the thickness of the plate in decimals of an inch, and call the product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2," and the quotient will give the percentage of strength of the rivets at the joint as compared with the solid plate.

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 2.5881 inches equal pitch from center to center of rivet holes in each row.

Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let 2 equal number of rows of rivet holes in lap.

Let 3125 ten thousandths of an inch equal thickness of plate.

Then we have, first the percentage of strength of plate at joint:

$$\frac{2.5881-.625}{2.5881}=.75+ \quad \text{Percentage of strength of plate at joint.}$$

Performing the operation, we have:

```
           2.5881  Pitch of rivet holes.
            .625   Diameter of rivet hole.
           ------
2.5881 ) 1.96310 (0.75+  Percentage of strength of plate at joint.
         1 81167
         -------
           151430
           129405
          -------
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Next we have the percentage of strength of the rivets at joint:

$$\frac{.625 \times .625 \times .7854 \times 2}{2.5881 \times .3125} = .75+ \quad \text{Percentage of strength of rivets at joint.}$$

Performing the operation, we have:

```
          .625   Diameter of rivet hole.
          .625   Diameter of rivet hole.
         -----
          3125
         1250
        3750
        ------
        .390625
          .7854
       --------
        1562500
       1953125
      3125000
     2734375
   ------------
   .3067968750  Area of diameter of rivet hole.
             2  Number of rows of rivets.
   ------------
   .6135937500  "Product No. 1."
```

Next, multiplying the pitch of rivets by the thickness of the plate, we have:

```
      2.5881  Pitch of rivets.
       .3125  Thickness of plate.
     -------
      129405
      51762
     25881
    77643
   ---------
   .80878125  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
.80878125 ) .6135937500 ( .75+  Percentage of strength of rivets at joint.
            566146875
            ----------
             474468750
             404390625
             ----------
```

STRENGTH OF TRIPLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF TRIPLE-RIVETED LAP JOINTS—IRON PLATES AND IRON RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch, and divide the remainder by the pitch, and the quotient will give the percentage of strength of the plate at joint.

Second, multiply the area of diameter of one rivet hole by the number of rows of rivets (3), and call the product "Product No. 1."

Third, multiply the pitch of rivets from center to center by the thickness of material in the plate, and call the product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2," and the quotient will give the percentage of strength of rivets at joint.

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 875 one thousandths of an inch ($\frac{7}{8}$ inch) equal diameter of rivet hole.

Let 3.2802 inches equal pitch of rivet holes from center to center.

Let 3 equal number of rows of rivet holes.

Let 75 one hundredths of an inch ($\frac{3}{4}$ inch) equal thickness of plate.

Then we have, first the percentage of strength of plate at joint:

$$\frac{3.2802-.875}{3.2802}=.7332+ \quad \text{Percentage of strength of plate at joint.}$$

Performing the operation, we have:

```
           3.2802  Pitch of rivet holes.
            .875   Diameter of rivet hole.
3.2802 ) 2.40520 ( 0.7332+  Percentage of strength of plate at joint.
         2 29614
           109060
            98406
            106540
             98406
              81340
              65604
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Next we have the percentage of strength of rivets at joint:

$$\frac{.875\times.875\times.7854\times3}{3.2802\times.75}=.7332+ \quad \text{Percentage of strength of rivets at joint.}$$

Performing the operation, we have:

```
         .875        Diameter of rivet hole.
         .875        Diameter of rivet hole.
        ------
        4375
       6125
      7000
      --------
      .765625
        .7854
     ----------
      3062500
     3828125
    6125000
   5359375
   ------------
   .6013218750       Area of diameter of rivet hole.
             3       Number of rows of rivets.
   ------------
  1.8039656250       "Product No. 1."
```

Next, multiplying the pitch of rivets by the thickness of the plate, we have:

```
   3.2802     Pitch of rivets.
      .75     Thickness of plate.
   -------
   164010
  2 29614
  --------
  2.460150    "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
2.46015) 1.803965625 (.7332+    Percentage of strength of rivets as compared with the solid plate.
         1 722105
         ---------
           818606
           738045
           -------
            805612
            738045
            -------
             675675
             492030
             ------
```

STRENGTH OF SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF SINGLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch and divide the remainder by the pitch, and the quotient will give the percentage of strength of plate at joint.

Second, square the diameter of rivet hole and multiply by .7854, which will give the area of cross section of rivet, and multiply the area by .8, and call the product "Product No. 1."

Third, multiply the pitch of rivets by the thickness of the plate, in decimals of an inch, and call the product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2," and the quotient will give the percentage of strength of rivets at joint.

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 1.60675 inches equal pitch of rivet holes.

Let 625 one thousandths of an inch equal diameter of rivet hole.

Let .8 equal a constant.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have first, percentage of strength of plate at joint:

$$\frac{1.60675-.625}{1.60675}=.61+ \quad \text{Percentage of strength of plate at joint.}$$

Performing the operation, we have:

$$\begin{array}{r l}
1.60675 & \text{Pitch of rivet holes.} \\
.625 & \text{Diameter of rivet hole.} \\
\hline
1.60675)\ .981750\ (0.61+ & \text{Percentage of strength of plate at joint.} \\
964050 & \\
\hline
177000 & \\
160675 & \\
\hline
\end{array}$$

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Next we have percentage of strength of rivets at joint:

$$\frac{.625\times.625\times.7854\times.8}{1.60675\times.25}=.61+ \quad \text{Percentage of strength of rivets at joint.}$$

Performing the operation, we have:

$$\begin{array}{r l}
.625 & \text{Diameter of rivet hole.} \\
.625 & \text{Diameter of rivet hole.} \\
\hline
3125 & \\
1250 & \\
3750 & \\
\hline
.390625 & \\
.7854 & \\
\hline
1562500 & \\
1953125 & \\
3125000 & \\
2734375 & \\
\hline
.3067968750 & \text{Area of diameter of rivet hole} \\
.8 & \text{A constant.} \\
\hline
.24543750000 & \text{"Product No. 1."}
\end{array}$$

Next, multiplying the pitch of rivets by the thickness of plate, we have:

```
  1.60675  Pitch of rivets.
      .25  Thickness of plate
 --------
   803375
  321350
 --------
 .4016875  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
.4016875) .24543750 (0.61+   Percentage of strength of riv-
           24101250               ets at joint.
           --------
             4425000
             4016875
             -------
```

STRENGTH OF DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

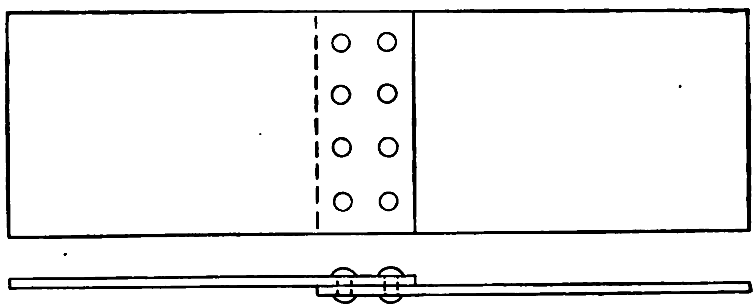


Fig. 63

TO DETERMINE THE PERCENTAGE OF STRENGTH OF DOUBLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—First, subtract the diameter of one rivet hole from the pitch, then divide the remainder by the pitch, and the quotient will give the percentage of strength of the plate at joint.

Second, multiply the area of diameter of one rivet hole by the number of rows of rivets (2), *then multiply the product by* .80, *and call the last product "Product No.* 1."

Third, multiply the pitch from center to center of rivets by the thickness of material in the plate in decimals of an inch, and call the product "Product No. 2."

Fourth, divide "Product No. 1" *by "Product No.* 2" *and the quotient will give the percentage of strength of the rivets at joint as compared with the solid plate.*

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 2.588 inches equal pitch from center to center of rivet holes in each row.

Let 625 one thousandths of an inch ($\frac{5}{8}$ inch) equal diameter of rivet hole.

Let 2 equal number of rows of rivet holes in a double-riveted joint.

Let .80 equal percentage of shearing strength of rivets as compared with tensile strength of plate.

Let 25 one hundredths of an inch equal thickness of plate.

Then we have first, the percentage of strength of plate at joint:

$$\frac{2.588-.625}{2.588}=.75+ \quad \text{Percentage of strength of plate at joint.}$$

Performing the operation in the ordinary way, we have:

```
        2.588  Pitch of rivet holes.
         .625  Diameter of rivet hole.
       ------
2.588) 1.9630 (0.75+  Percentage of strength of plate at joint.
       1 8116
       ------
        15140
        12940
        -----
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Next, we have the percentage of strength of the rivets at joint:

$$\frac{.625\times.625\times.7854\times2\times.80}{2.588\times.25}=.75+ \quad \text{Percentage of strength of rivets at joint.}$$

Performing the operation in the ordinary way, we have:

```
          .625  Diameter of rivet hole.
          .625  Diameter of rivet hole.
          ----
          3125
         1250
        3750
        ------
       .390625
         .7854
       -------
       1562500
      1953125
     3125000
    2734375
    -----------
```

Am't carried forward, .3067968750 Area of diameter of rivet hole.

Am't brought forward,	.3067968750	Area of diameter of rivet hole.
	2	Number of rows of rivets.
	.6135937500	
	.80	Percentage of shearing strength of rivets.
	.490875000000	"Product No. 1."

Then, multiplying the pitch of rivets by the thickness of the plate, we have:

2.588	Pitch of rivets.
.25	Thickness of plate.
12940	
5176	
.64700	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

.64700).490875 (0.75+	Percentage of strength of rivets at joint.
452900	
379750	
323500	

It will be observed that the diameter of the rivet and rivet hole have been taken as being the same in the foregoing examples. The reason is that while the rivet before riveting is always smaller than the rivet hole, but after riveting, if the work is properly done, the rivet will fill the hole, and will therefore be equal in diameter; hence the strength of riveted joints is determined by the strength of the joints after completion.

STRENGTH OF TRIPLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

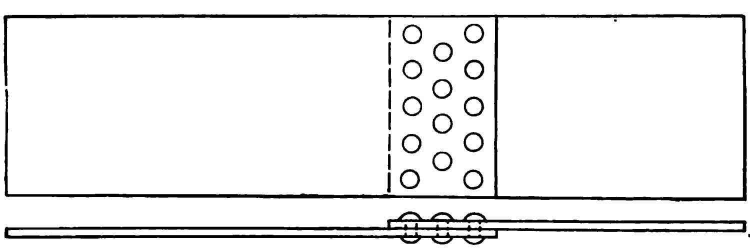


Fig. 64

TO DETERMINE THE PERCENTAGE OF STRENGTH OF TRIPLE-RIVETED LAP JOINTS—STEEL PLATES AND STEEL RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch, then divide the remainder by the pitch, and the quotient will give the percentage of strength of the plate at joint.

Second, multiply the area of diameter of one rivet hole by the number of rows of rivets (3), *then multiply the product by* .80, *and call the last product "Product No.* 1."

Third, multiply the pitch of rivets from center to center by the thickness of material in the plate, and call the product "Product No. 2."

Fourth, divide "Product No. 1" *by "Product No.* 2" *and the quotient will give the percentage of strength of the rivets at joint.*

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 2.87 inches equal pitch of rivet holes.

Let 75 one hundredths of an inch equal diameter of rivet hole.

Then we have:

$$\frac{2.87-.75}{2.87}=.73+ \text{ Percentage of strength of plate at joint}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl}
2.87 & \text{Pitch of rivet holes.} \\
.75 & \text{Diameter of rivet hole.} \\
\hline
2.87\,)\,2.120\,(0.73+ & \text{Percentage of strength of plate at joint.} \\
2\,009 & \\
\hline
1110 & \\
861 & \\
\hline
\end{array}$$

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.

Let 3 equal number of rows of rivets.

Let .80 equal percentage of shearing strength of rivets as compared with the tensile strength of plate.

Let 2.87 inches equal pitch of rivets from center to center.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75\times.75\times.7854\times3\times.80}{2.87\times.5}=.73+ \text{ Percentage of strength of rivets at joint.}$$

Performing the operation in the ordinary way, we have:

.75	Diameter of rivet hole.
.75	Diameter of rivet hole.
375	
525	
.5625	
.7854	
22500	
28125	
45000	
39375	
.44178750	Area of diameter of rivet hole.
3	Number of rows of rivets.
1.32536250	
.80	Percentage of shearing strength of rivets as compared with tensile strength of plate.
1.0602900000	"Product No. 1."

Next, multiplying the pitch from center to center of rivets by the thickness of plate, we have:

2.87	Pitch of rivets.
.5	Thickness of plate.
1.435	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

1.435) 1.06029 (.73+	Percentage of strength of rivets at joint.
1 0045	
5579	
4305	

BUTT-STRAP JOINTS.

Where single butt-straps are employed in butt joints the thickness of the strap must not be less than the thickness of the plate; where double butt-straps are employed the thickness of the straps must not be less than five-eighths of the thickness of the plate.

Where single butt-straps are employed, the rule for determining the pitch for single, double, or triple riveting, and for determining the percentage of strength at the joints, is the same as for riveted lap joints. But for double butt-strapping the rule is different, on account of the rivets being in double shear instead of single shear, as in the case of lap joints or butt joints with single straps. In lap joints and single butt joints the rivets are subjected to being sheared through one cross section of the rivets, while in double butt-strapping the rivets are subjected to being sheared through two cross sections of the rivets. Hence, the difference between the rules for lap and single butt-strap joints and double butt-strap joints.

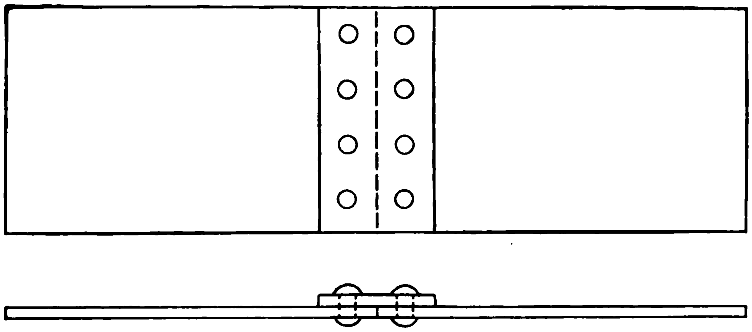


Fig. 65

SINGLE-RIVETED BUTT JOINT, WITH SINGLE BUTT-STRAP.

PITCH OF RIVET HOLES FOR SINGLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

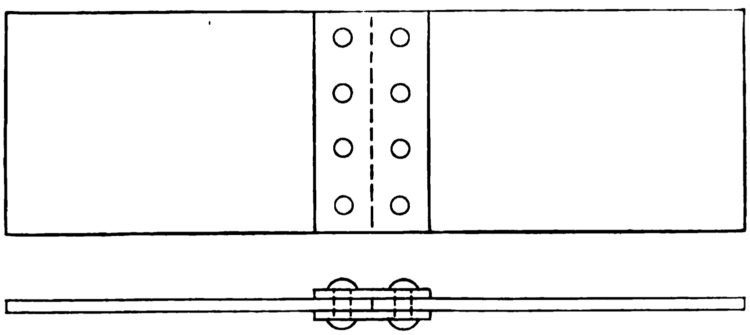


Fig. 66

SINGLE-RIVETED BUTT JOINT, WITH DOUBLE BUTT-STRAP.

TO DETERMINE THE PITCH OF RIVET HOLES FOR SINGLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, multiply the area of the diameter of rivet hole by 1.75, *then multiply the product by* .80, *and call the product "Product No.* 1."

Second, divide "Product No. 1" *by the thickness of plate, and add the diameter of rivet hole to the quotient, and the sum will give the required pitch in inches from center to center of rivets.*

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.

Let 1.75 equal a constant for rivets in double shear.

Let .80 equal percentage of shearing strength of rivets.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75\times.75\times.7854\times1.75\times.80}{.5}+.75=1.987+\text{ inches.}$$

Pitch of rivet holes from center to center.

Performing the operation in the ordinary way, we have:

```
          .75   Diameter of rivet hole.
          .75   Diameter of rivet hole.
          ---
          375
         525
        -----
        .5625
        .7854
        -----
        22500
       28125
      45000
     39375
    ---------
    .44178750   Area of diameter of rivet hole.
         1.75   A constant for rivets in double shear.
   ----------
    220893750
   309251250
  44178750
 ------------
 .7731281250
          .80   Percentage of shearing strength of rivets.
 ------------
.618502500000   "Product No. 1."
```

Next, dividing "Product No. 1" by the thickness of the material in the plate, and adding the diameter of rivet hole to the quotient, we have:

```
.5) .6185
    -----
    1.237
     .75   Diameter of rivet hole.
    -----
    1.987 inches.  Pitch of rivet holes from center to center.
```

PITCH OF RIVET HOLES FOR DOUBLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

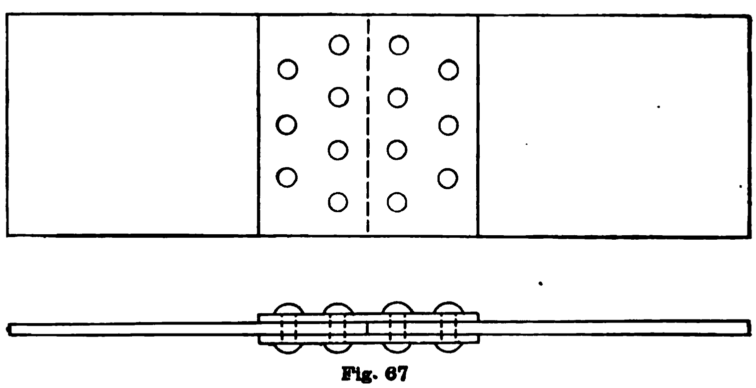


Fig. 67

STAGGERED DOUBLE-RIVETED BUTT JOINT, WITH DOUBLE BUTT-STRAP.

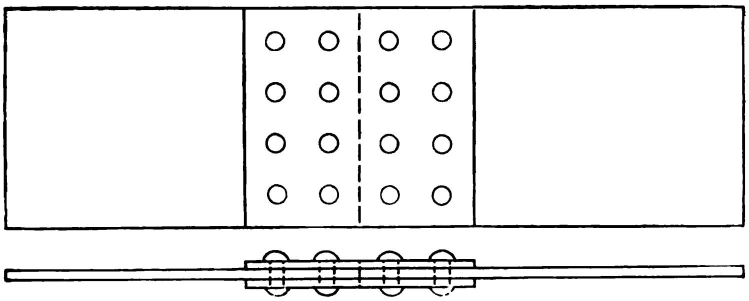


Fig. 68

CHAIN DOUBLE-RIVETED BUTT JOINT, WITH DOUBLE BUTT-STRAP.

TO DETERMINE THE PITCH OF RIVET HOLES FOR DOUBLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, multiply the area of diameter of rivet hole by the number of rows of rivet holes (2), *then multiply the product by* 1.75, *on account of the rivets being in double shear; then multiply the last product by* .80, *and call the resulting product "Product No.* 1."

Second, divide "Product No. 1" *by the thickness of the plate and add the diameter of rivet hole to the quotient, and the sum will give the pitch of rivet holes from center to center in inches.*

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.

Let 2 equal number of rows of rivets.

Let 1.75 equal a constant for rivets in double shear.

Let .80 equal percentage of shearing strength of rivets.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75\times.75\times.7854\times2\times1.75\times.80}{.5}+.75=3.22+\text{ inches.}$$ Pitch of rivet holes from center to center.

Performing the operation in the ordinary way, we have:

.75	Diameter of rivet hole.
.75	Diameter of rivet hole.
375	
525	
.5625	
.7854	
22500	
28125	
45000	
39375	
.44178750	Area of diameter of rivet hole.

Am't carried forward, .44178750

Am't brought forward,	.44178750	Area of diameter of rivet hole.
	2	Number of rows of rivet holes.
	.88357500	
	1.75	Constant for rivets in double shear.
	441787500	
	618502500	
	88357500	
	1.5462562500	
	.80	Percentage of shearing strength of rivets.
	1.237005000000	"Product No. 1."

Next, dividing "Product No. 1" by the thickness of material in the plate and adding the diameter of rivet hole to the quotient, we have:

.5) 1.237005	
2.47401	
.75	Diameter of rivet hole.
3.22401 inches.	Pitch of rivet holes from center to center.

PITCH OF RIVET HOLES FOR TRIPLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATE, STEEL BUTT-STRAPS AND STEEL RIVETS.

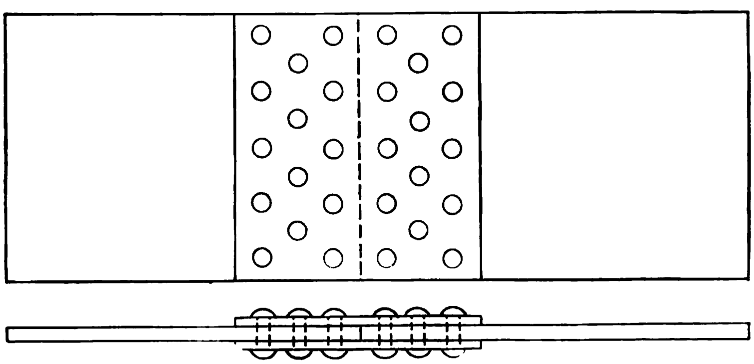


Fig. 69

TRIPLE-RIVETED BUTT-STRAP JOINT, WITH DOUBLE BUTT-STRAP.

NOTE.—Frequently such joints have two rows of rivets, regularly spaced, and one row with double the space between the rivets from center to center; therefore, instead of using 3 as a multiplier we use 2.5 to represent the rows of rivets where the third row is double spaced.

TO DETERMINE THE PITCH OF RIVET HOLES FOR TRIPLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, multiply the area of diameter of rivet hole by 2.5, *then multiply the product by* 1.75, *on account of rivets being in double shear; then multiply the last product by* .80, *and call the resulting product "Product No.* 1."

Second, divide "Product No. 1" by the thickness of the plate (not the straps), and add the diameter of rivet hole to the quotient, and the sum will give the pitch of rivet holes from center to center in inches.

Example.—Let 1.3416 inches equal diameter of rivet hole.
Let 2.5 equal number of rows of rivets.
Let 1.75 equal a co-efficient for rivets in double shear.
Let .80 equal percentage of shearing strength of rivets as compared with tensile strength of plate.
Let 1.25 inches equal thickness of plate.

Then we have:

$$\frac{1.3416\times1.3416\times.7854\times2.5\times1.75\times.80}{1.25}+1.3416=5.2997+\text{inches.}$$

Required pitch of rivet holes from center to center.

Performing the operation in the ordinary way, we have:

1.3416 Diameter of rivet hole.
1.3416 Diameter of rivet hole.

80496
13416
53664
40248
1 3416

1.79989056
.7854

719956224
899945280
1439912448
1 259923392

1.413634045824 Area of diameter of rivet hole.
2.5 Representing rows of rivets.

7068170229120
2 827268091648

3.534085114560 0
1.75 A constant for rivets in double shear.

176704255728000
2 47385958019200
3 5340851145600

6.1846489504800 00
.80 Percentage of shearing strength of rivets.

4.9477191603840000 "Product No. 1."

Next, dividing "Product No. 1" by the thickness of the plate, and adding the diameters of rivet hole to the quotient, we have:

```
1.25) 4.947719 (3.9581
      3 75      1.3416  Diameter of rivet hole.
      ----      ------
      1 197     5.2997 inches. Required pitch of rivet holes from center to center.
      1 125
      -----
        727
        625
       ----
       1021
       1000
       ----
         219
         125
        ----
```

The pitch, as shown in the example, is the required pitch for two rows of rivets, while the pitch in the third row is to be double that in the double rows.

STRENGTH OF SINGLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF SINGLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, subtract the diameter of the rivet hole from the pitch, then divide the remainder by the pitch, and the quotient will give the percentage of strength of the plate at joint.

Second, multiply the area of diameter of rivet hole by 1.75, *on account of rivets being in double shear, then multiply the product by* .80, *and call the last product "Product No.* 1."

Third, multiply the pitch from center to center of rivets by the thickness of the plate, and call the product "Product No. 2."

Fourth, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the percentage of strength of rivets at joint.*

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 1.987 inches equal pitch of rivet holes from center to center.

Let 75 one hundredths of an inch equal diameter of rivet hole.

Then we have:

$$\frac{1.987-.75}{1.987}=.622+ \quad \text{Percentage of strength of plate at joint.}$$

Performing the operation in the ordinary way, we have:

```
            1.987  Pitch of rivet holes.
             .75   Diameter of rivet hole.
           ------
  1.987) 1.2370 (0.622+  Percentage of strength of plate
         1 1922                at joint.
         ------
           4480
           3974
           ----
            5060
            3974
            ----
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Example.—Let 75 one hundredths of an inch equal diameter of rivet.

Let 1.75 equal a constant for rivets in double shear.

Let .80 equal percentage of shearing strength of rivets.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75 \times .75 \times .7854 \times 1.75 \times .80}{1.987 \times .5} = .622+$$ Percentage of strength of rivets at joints.

Performing the operation in the ordinary way, we have:

```
              .75   Diameter of rivet hole.
              .75   Diameter of rivet hole.
             ----
              375
             525
             ----
            .5625
            .7854
            -----
            22500
           28125
          45000
         39375
         --------
        .44178750   Area of diameter of rivet hole.
             1.75   Constant for rivets in double shear.
        ---------
        220893750
      309251250
      44178750
      -----------
     .7731281250
              .80   Percentage of shearing strength of
     ------------         rivets.
    .618502500000   "Product No. 1."
```

Next, multiplying the pitch of rivets from center to center by the thickness of the plate, we have:

1.987	Pitch of rivets.
.5	Thickness of plate.
.9935	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

.9935).6185025 (.622+ Percentage of strength of rivets at joint.
59610
22402
19870
25325
19870

STRENGTH OF DOUBLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF DOUBLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch, then divide the remainder by the pitch, and the quotient will give the percentage of strength of plate at joint.

Second, multiply the area of diameter of rivet hole by the number of rows of rivets (2); *then multiply the product by* 1.75, *on account of rivets being in double shear; then multiply the last product by* .80, *and call the resulting product "Product No.* 1."

Third, multiply the pitch of the rivets from center to center by the thickness of the plate, and call the product "Product No. 2."

Fourth, divide "Product No. 1" *by "Product No.* 2" *and the quotient will give the percentage of strength of the rivets at joint.*

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 3.22 inches equal pitch of rivet holes.
Let 75 one hundredths of an inch equal diameter of rivet hole.

Then we have:

$$\frac{3.22-.75}{3.22} = .76+$$ Percentage of strength of plate at joint.

Performing the operation in the ordinary way, we have:

3.22	Pitch of rivet holes.
.75	Diameter of rivet hole.

3.22) 2.470 (0.76+ Percentage of strength of plate at joint.
2 254
2160
1932

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Example.—Let 75 one hundredths of an inch equal diameter of rivet hole.

Let 2 equal number of rows of rivets.

Let 1.75 equal a constant for rivets in double shear.

Let .80 equal percentage of shearing strength of rivets.

Let 3.22 inches equal pitch of rivets from center to center.

Let 5 tenths of an inch equal thickness of plate.

Then we have:

$$\frac{.75\times.75\times.7854\times2\times1.75\times.80}{3.22\times.5}=.76+$$ Percentage of strength of rivets at joint.

Performing the operation in the ordinary way, we have:

```
          .75  Diameter of rivet hole.
          .75  Diameter of rivet hole.
         ----
          375
         525
         ----
         .5625
         .7854
        ------
         22500
        28125
       45000
      39375
    ----------
     .44178750  Area of diameter of rivet hole.
             2
    ----------
     .88357500
          1.75  Constant for rivets in double shear.
    ----------
     441787500
    618502500
   88357500
  ------------
  1.5462562500
           .80  Percentage of shearing strength of rivets.
  --------------
1.237005000000  "Product No. 1."
```

Next, multiplying the pitch of rivets from center to center by the thickness of plate, we have:

```
 3.22  Pitch of rivets.
   .5  Thickness of plate.
 -----
1.610  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
1.610) 1.237005 (.76+  Percentage of strength of rivets at joint.
       1 1270
       ------
         11005
          9660
         -----
```

STRENGTH OF TRIPLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

TO DETERMINE THE PERCENTAGE OF STRENGTH OF TRIPLE-RIVETED DOUBLE BUTT-STRAP JOINTS—STEEL PLATES, STEEL BUTT-STRAPS AND STEEL RIVETS.

RULE.—First, subtract the diameter of rivet hole from the pitch, and divide the remainder by the pitch, the quotient will give the percentage of strength of plate at joint.

Second, multiply the area of diameter of rivet hole by 2.5, *then multiply the product by* 1.75, *and multiply the last product by* .80, *and call the resulting product "Product No. 1."*

Third, multiply the pitch of rivets from center to center in inches, by the thickness of the plate in decimals of an inch, and call the product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2," and the quotient will give the percentage of strength of rivets at joint.

PERCENTAGE OF STRENGTH OF PLATE AT JOINT.

Example.—Let 5.2997 inches equal pitch of rivet holes from center to center.

Let 1.3416 inches equal diameter of rivet hole.

Then we have:

$$\frac{5.2997-1.3416}{5.2997}=.74+ \text{ Percentage of strength of plate at joint.}$$

Performing the operation in the ordinary way, we have:

```
             5.2997  Pitch of rivet holes.
             1.3416  Diameter of rivet hole.
            --------
5.2997 ) 3.95810 (0.74+  Percentage of strength of plate at joint.
         3 70979
         -------
          248310
          211988
          ------
```

PERCENTAGE OF STRENGTH OF RIVETS AT JOINT.

Example.—Let 1.3416 inches equal diameter of rivet hole.

Let 2.5 equal number of rows of rivets.

Let 1.75 equal a co-efficient for rivets in double shear.

Let .80 equal percentage of shearing strength of rivets as compared with tensile strength of plate.

Let 5.2997 inches equal pitch of rivets from center to center.

Let 1.25 inches equal thickness of plate.

Then we have:

$$\frac{1.3416\times1.3416\times.7854\times2.5\times1.75\times.80}{5.2997\times1.25}=.74+$$ Percentage of strength of rivets at joint.

Performing the operation in the ordinary way, we have:

1.3416 Diameter of rivet hole.
1.3416 Diameter of rivet hole.

80496
13416
53664
40248
1 3416

1.79989056
.7854

719956224
899945280
1439912448
1 259923392

1.413634045824 Area of diameter of rivet hole.
2.5 Number of rows of rivets.

7068170229120
2 827268091648

3.5340851145600
1.75 A co-efficient for rivets in double shear.

176704255728000
2 47385958019200
3 53408511456000

6.184648950480000
.80 Percentage of shearing strength of rivets to tensile strengh of plate.

4.94771916038400000 "Product No. 1."

Next, multiplying the pitch by the thickness of plate, we have:

5.2997 inches. Pitch.
1.25 inches. Thickness of plate.

264985
105994
52997

6.624625 "Product No. 2."

Next, dividing "Product No. 1" by Product No. 2," we have.

6.624625) 4.94771916 (.74+ Percentage of strength of rivets at joint.
4 6372375

31048166
26498500

TABLE OF RIVETED JOINTS.

IRON PLATES, IRON RIVETS, AND HOLES DRILLED.

SINGLE RIVETED JOINTS.

Thickness of Plate.	Diameter of Rivet Holes.	Distance of Center of Rivet Holes from Edge of Sheet.	Pitch of Rivet Holes from Center to Center.	Breadth of Lap.	Percent'ge of Strength of Plate at the Joint.	Percent'ge of Strength of Rivets at the Joint.
1/4 in.	5/8 in.	15/16 in.	1.8521 in.	1 7/8 in.	66.2	66.2
5/16 "	11/16 "	1 1/32 "	1.8754 "	2 1/16 "	63.3	63.3
3/8 "	3/4 "	1 1/8 "	1.9281 "	2 1/4 "	61.1	61.1
7/16 "	13/16 "	1 7/32 "	1.9976 "	2 7/16 "	59.3	59.3
1/2 "	7/8 "	1 5/16 "	2.0776 "	2 5/8 "	57.8	57.8
9/16 "	15/16 "	1 13/32 "	2.1646 "	2 13/16 "	56.6	56.6
5/8 "	1 "	1 1/2 "	2.2566 "	3 "	55.6	55.6
11/16 "	1 1/16 "	1 19/32 "	2.3521 "	3 3/16 "	54.8	54.8
3/4 "	1 1/8 "	1 11/16 "	2.4503 "	3 3/8 "	54.0	54.0

DOUBLE RIVETED JOINTS.

Thickness of Plate.	Diameter of Rivet Holes.	Distance of Center of Outer Row of Rivet Holes from Edge of Sheet.	Pitch of Rivet Holes from Center to Center.	Distance between Centers of Rows of Rivet Holes for Chain Riveting.	Distance between Centers of Rows of Rivet Holes for Zigzag Riveting.	Breadth of Lap for Chain Riveting.	Breadth of Lap for Zigzag Riveting.	Percent'ge of Strength of Plate at the Joint.	Percent'ge of Strength of Rivets at the Joint.
1/4 in.	9/16 in.	27/32 in.	2.5505 in.	1 7/16 in.	1 3/16 in.	3 1/8 in.	2 7/8 in.	77.9	77.9
5/16 "	5/8 "	15/16 "	2.5885 "	1 9/16 "	1 1/4 "	3 7/16 "	3 1/8 "	75.8	75.8
3/8 "	11/16 "	1 1/32 "	2.6674 "	1 3/4 "	1 5/16 "	3 13/16 "	3 3/8 "	74.2	74.2
7/16 "	3/4 "	1 1/8 "	2.7696 "	1 7/8 "	1 3/8 "	4 1/8 "	3 5/8 "	72.9	72.9
1/2 "	13/16 "	1 7/32 "	2.8864 "	2 "	1 7/16 "	4 7/16 "	3 7/8 "	71.8	71.8
9/16 "	7/8 "	1 5/16 "	3.0130 "	2 3/16 "	1 1/2 "	4 3/4 "	4 1/8 "	70.9	70.9
5/8 "	15/16 "	1 13/32 "	3.1464 "	2 5/16 "	1 5/8 "	5 1/8 "	4 3/8 "	70.2	70.2
11/16 "	1 "	1 1/2 "	3.2848 "	2 1/2 "	1 11/16 "	5 1/2 "	4 5/8 "	69.5	69.5
3/4 "	1 1/16 "	1 19/32 "	3.4268 "	2 5/8 "	1 13/16 "	5 13/16 "	5 "	68.9	68.9

STEEL PLATES, STEEL RIVETS, AND HOLES DRILLED.

SINGLE RIVETED JOINTS.

Thickness of Plate.	Diameter of Rivet Holes.	Distance of Center of Rivet Holes from Edge of Sheet.	Pitch of Rivet Holes from Center to Center.	Breadth of Lap.	Percentage of Strength of Plate at the Joint.	Percentage of Strength of Rivets at the Joint.
1/4 in.	11/16 in.	1 1/32 in.	1.8754 in.	2 1/16 in.	63.3	63.3
5/16 "	3/4 "	1 1/8 "	1.8805 "	2 1/4 "	60.1	60.1
3/8 "	13/16 "	1 7/32 "	1.9180 "	2 7/16 "	57.6	57.6
7/16 "	7/8 "	1 5/16 "	1.9745 "	2 5/8 "	55.6	55.6
1/2 "	15/16 "	1 13/32 "	2.0419 "	2 13/16 "	54.8	54.8
9/16 "	1 "	1 1/2 "	2.1170 "	3 "	52.7	52.7
5/8 "	1 1/16 "	1 19/32 "	2.1974 "	3 3/16 "	51.6	51.6
11/16 "	1 1/8 "	1 11/16 "	2.2816 "	3 3/8 "	50.6	50.6
3/4 "	1 3/16 "	1 25/32 "	2.3688 "	3 9/16 "	49.8	49.8

DOUBLE RIVETED JOINTS.

Thickness of Plate.	Diameter of Rivet Holes.	Distance of Center of Outer Row of Rivet Holes from Edge of Sheet.	Pitch of Rivet Holes from Center to Center.	Distance between Centers of Rows of Rivet Holes for Chain Riveting.	Distance between Centers of Rows of Rivet Holes for Zigzag Riveting.	Breadth of Lap for Chain Riveting.	Breadth of Lap for Zigzag Riveting.	Percentage of Strength of Plate at the Joint.	Percentage of Strength of Rivets at the Joint.
1/4 in.	5/8 in.	15/16 in.	2.5885 in.	1 5/8 in.	1 1/4 in.	3 1/2 in.	3 1/8 in.	75.8	75.8
5/16 "	11/16 "	1 1/32 "	2.5881 "	1 3/4 "	1 5/16 "	3 13/16 "	3 3/8 "	73.4	73.4
3/8 "	3/4 "	1 1/8 "	2.6349 "	1 7/8 "	1 3/8 "	4 1/8 "	3 5/8 "	71.5	71.5
7/16 "	13/16 "	1 7/32 "	2.7084 "	2 "	1 13/32 "	4 7/16 "	3 7/8 "	70.0	70.0
1/2 "	7/8 "	1 5/16 "	2.7992 "	2 1/8 "	1 1/2 "	4 3/4 "	4 1/8 "	68.7	68.7
9/16 "	15/16 "	1 13/32 "	2.9010 "	2 1/4 "	1 9/16 "	5 1/16 "	4 3/8 "	67.6	67.6
5/8 "	1 "	1 1/2 "	3.0106 "	2 3/8 "	1 5/8 "	5 3/8 "	4 5/8 "	66.7	66.7
11/16 "	1 1/16 "	1 19/32 "	3.1259 "	2 1/2 "	1 11/16 "	5 11/16 "	4 7/8 "	66.0	66.0
3/4 "	1 1/8 "	1 11/16 "	3.2455 "	2 5/8 "	1 3/4 "	6 "	5 1/8 "	65.3	65.3

CHAPTER VII.

BOILER FLUES.

There is no question connected with the science of steam engineering so little understood by engineers generally, as that of boiler flues. Until the experiments of Sir WILLIAM FAIRBAIRN demonstrated that a flue, 5 feet in length, of a given diameter and given thickness of material, would stand just double the pressure per square inch to collapse it that a flue 10 feet in length, of the same given diameter and same given thickness of material, would stand. Hence, in making calculations in regard to the safe-working or collapsing pressure of boiler flues, the length of the flue must be taken into consideration as well as the thickness of its material and its diameter. Therefore, as the length of the flue is increased the flue becomes proportionately weaker. In this respect it differs very materially from the shell of a boiler with its internal pressure. The power of the flue's resistance to collapse depends, in a great measure, upon the circularity of its form; for the reason that as the flue deviates in form from that of a perfect circle it becomes proportionately weak and less able to resist collapse; hence, the importance of making all boiler flues to conform as near as possible to that of a perfect circle. Carelessness in these matters has caused many serious accidents and the destruction of many human lives.

The practice of making large boiler flues with the same thickness of material as that employed in making the shells of the boilers, has been another fruitful source of a large number of boiler accidents. And yet, the lack of information upon this subject among engineers and boiler makers is not to be wondered at, when the United States government itself had taken no notice of the length of boiler flues 16 inches and less in diameter, nor even the pressure of steam in that direction, until a few years ago. The only rule upon the subject, so far as flues 16 inches and less in diameter were concerned, was, "that all flues having a diameter of 16 inches shall have a thickness of $\frac{5}{16}$ of an inch, and in proportion for greater or less diameters." And this was without any regard to the length of the flue or to the pressure the flue was required to carry. To the credit of the government, however, it can be said that that error has been corrected, and most excellent rules governing such flues have been adopted.

Many rules relating to boiler flues have been promulgated by different authorities, and many of those rules are based entirely upon theory. It is, therefore, not always safe to follow such rules. The best teacher in all such matters is practice. Therefore, all of the rules that will be given here are such as are based upon actual experience and have been found safe in practice. It is an easy matter to make a flue that will be perfectly safe; but aside from the question of safety, the question of efficiency in the generation of steam must not be lost sight of. Therefore, while the flue may be made heavy enough to withstand any given pressure, yet its efficiency may be entirely destroyed. It is well known that thin material is a better conductor of heat than thick material. And here again is the danger of losing sight of the important question of safety. The student is, therefore, informed that the rules here laid down, are laid down with two important objects in view—safety and efficiency. And the conclusions herein reached are based entirely upon practical experience, so that the two things most essential—safety and efficiency—are kept constantly in view.

SAFE-WORKING PRESSURE OF FLUES.

TO DETERMINE THE SAFE-WORKING PRESSURE OF LAP-WELDED BOILER FLUES, NOT MADE IN SECTIONS.

RULE.—First, square the thickness of material in hundredths of an inch, and then multiply the product by the constant whole number 806300, *and call the last product "Product No. 1."*

Second, multiply the diameter of the flue in inches by the length of the flue in feet, and then multiply the product by 3, *and call the last product "Product No. 2."*

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the safe-working pressure for a flue made in one length.

Example.—Let 35 one hundredths of an inch equal thickness of material.

Let 806300 equal a constant.

Let 16 inches equal diameter of the flue.

Let 25 feet equal length of the flue.

Let 3 equal a constant.

Then we have:

$$\frac{.35\times.35\times806300}{16\times25\times3}=82.31-\text{ lbs. Safe-working pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

```
.35×.35=.1225  The square of the thickness of material.
      806300  A constant.
      -------
      36 7500
     735 0
    9800
    -----------
    98771.7500  "Product No. 1."

        16  Diameter of the flue in inches.
        25  Length of the flue in feet.
       ---
        80
       32
       ---
       400
         3  A constant.
      ----
      1200  "Product No. 2.
```

Finally, dividing "Product No. 1" by "Product No. 2." we have:

```
1200) 98771.75 (82.31— lbs.
      9600
      ----
       2771
       2400
       ----
        3717
        3600
        ----
         1175
         1200
         ----
```

THICKNESS OF MATERIAL OF FLUES.

TO DETERMINE THE THICKNESS OF MATERIAL REQUIRED FOR ANY GIVEN PRESSURE, GIVEN DIAMETER AND GIVEN LENGTH OF FLUE, NOT MADE IN SECTIONS.

RULE.—First, multiply the diameter of the flue in inches, by the length of the flue in feet, then multiply the product by the constant 3, *and call the last product "Product No.* 1."

Second, multiply "Product No. 1" *by the given pressure per square inch, and call the product "Product No.* 2."

Third, divide "Product No. 2" *by the constant whole number* 806300, *and extract the square root of the quotient, the answer will give the thickness of material required in decimals of an inch.*

Example.—Let 16 inches equal diameter of the flue.
Let 25 feet equal length of the flue.
Let 3 equal a constant.
Let 82.31 pounds equal given pressure per square inch.
Let 806300 equal a constant.

Then we have:

$$\sqrt{\frac{16\times25\times3\times82.31}{806300}}=.35 \text{ Decimals of an inch. Thickness of material required.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl}
16 & \text{Diameter of flue.} \\
\underline{25} & \text{Length of flue.} \\
80 & \\
\underline{32} & \\
400 & \\
\underline{3} & \text{A constant.} \\
1200 & \text{“ Product No. 1.”} \\
\underline{82.31} & \text{Pressure per square inch.} \\
12\,00 & \\
360\,0 & \\
2400 & \\
\underline{9600} & \\
98772.00 & \text{“ Product No. 2.”}
\end{array}$$

Dividing “Product No. 2” by 806300, we have:

$$\begin{array}{l}
806300\,)\,98772.00\,(\,0.1225+ \quad \text{Square of thickness of material.} \\
\underline{80630\,0} \\
1814200 \\
\underline{1612600} \\
2016000 \\
\underline{1612600} \\
4034000 \\
\underline{4031500}
\end{array}$$

Extracting the square root of the quotient, we have:

$$\begin{array}{rl}
 & .1\dot{2}2\dot{5}\,(.35 \quad \text{Thickness of required material in hundredths of an inch.} \\
 & \underline{9} \\
65\,) & 325 \\
 & \underline{325}
\end{array}$$

DIAMETER OF FLUES.

TO DETERMINE THE DIAMETER REQUIRED FOR A FLUE NOT MADE IN SECTIONS, OF A GIVEN THICKNESS OF MATERIAL AND GIVEN LENGTH, TO CARRY A GIVEN SAFE-WORKING PRESSURE PER SQUARE INCH.

RULE.—First, multiply the square of the thickness of material in hundredths of an inch, by the constant whole number 806300, *and call the product "Product No.* 1."

Second, multiply the length of the flue in feet, by the constant 3, *and then multiply the product by the given safe-working pressure per square inch, and call the last product "Product No.* 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the required diameter of the flue in inches.*

Example.—Let 35 one hundredths of an inch equal thickness of material.

Let 806300 equal a constant.

Let 25 feet equal length of flue.

Let 3 equal a constant.

Let 82.31 pounds per square inch equal given pressure.

Then we have:

$$\frac{.35\times.35\times806300}{25\times3\times82.31}=16\text{— inches. Diameter required.}$$

Performing the operation in the ordinary way, we have:

```
        .35      Thickness of material.
        .35      Thickness of material.
       -----
        175
       105
       -----
       .1225     Square of the thickness of material.
       806300    A constant.
       ------
       36 7500
      735 0
     9800
    -----------
    98771.7500   "Product No. 1."
```

Next we have:

```
        25     Length of the flue.
         3     A constant.
       ----
        75
     82.31     Given pressure per square inch.
    ------
    411 55
   5761 7
   -------
   6173.25     "Product No. 2."
```

Then we have, "Product No. 1" divided by "Product No. 2:"

```
6173.25 ) 98771.75 ( 16— inches. The required diameter.
          617325
          -------
          3703925
          3703950
          -------
```

LENGTH OF FLUES.

TO DETERMINE THE LENGTH REQUIRED FOR A FLUE NOT MADE IN SECTIONS, OF A GIVEN THICKNESS OF MATERIAL AND GIVEN DIAMETER, TO CARRY A REQUIRED WORKING PRESSURE PER SQUARE INCH.

RULE.—First, multiply the square of the given thickness of material in hundredths of an inch by the constant whole number 806300, *and call the product "Product No.* 1."

Second, multiply the given diameter of the flue in inches by the constant 3, *and then multiply the product by the given pressure per square inch, and call the last product "Product No.* 2."

Third, divide "Product No. 1" *by "Product No.* 2" *and the quotient will give the required length of the flue in feet.*

Example.—Let 35 one hundredths of an inch equal thickness of material.

Let 806300 equal a constant.

Let 16 inches equal diameter of the flue.

Let 3 equal a constant.

Let 82.31 pounds per square inch equal working pressure.

Then we have:

$$\frac{.35\times.35\times806300}{16\times3\times82.31}=25\text{— feet. Required length of the flue.}$$

Next we have:

```
       .35    Thickness of material.
       .35    Thickness of material.
      ----
       175
      105
     -----
     .1225    Square of thickness of material.
    806300    A constant.
   -------
   36 7500
   735 0
  9800
------------
98771.7500    "Product No. 1."
```

Performing the operation in the ordinary way, we have:

```
    16  Diameter of flue.
     3  A constant.
  ----
    48
  82.31
 ------
 658 48
3292 4
-------
3950.88  "Product No. 2."
```

Then we have, "Product No. 1" divided by "Product No. 2:"

```
3950.88) 98771.75 (25— feet.  Required length of the flue.
         79017 6
         --------
          19754 15
          19754 40
          --------
```

COLLAPSING PRESSURE OF FLUES.

TO DETERMINE THE COLLAPSING PRESSURE OF FLUES NOT MADE IN SECTIONS.

RULE.—First, multiply the square of the thickness of material in the flue, in hundredths of an inch, by the constant whole number 806300, and call the product "Product No. 1."

Second, multiply the diameter of the flue, in inches by the length of the flue, in feet, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the collapsing pressure per square inch.

Example.—Let 35 one hundredths of an inch equal thickness of material.
Let 806300 equal a constant.
Let 16 inches equal diameter of the flue.
Let 25 feet equal length of the flue.

Then we have:

$$\frac{.35\times.35\times806300}{16\times25}=446.92+\text{ lbs.}$$ Collapsing pressure per square inch.

Performing the operation in the ordinary way, we have:

```
 .35  Thickness of material.
 .35  Thickness of material.
 ----
  175
 105
 -----
.1225  Square of thickness of material.
```

Am't carried forward, .1225

Am't brought forward,

```
      .1225      Square of thickness of material.
    806300       A constant.
    ------
    36 7500
   735 0
  9800
  -----------
  98771.7500
```

Next we have:

```
   16   Diameter of flue.
   25   Length of flue.
   --
   80
  32
  ---
  400   "Product No. 2."
```

Then we have, "Product No. 2" divided by "Product No. 1:"

```
400) 98771.75 (246.92+ lbs.   Pressure per square
     800                        inch required to
     ----                       collapse the flue.
     1877
     1600
     ----
      2771
      2400
      ----
       3717
       3600
       ----
        1175
         800
        ----
```

CHAPTER VIII.

PUMPS.

HORSE POWER REQUIRED TO ELEVATE WATER.

TO DETERMINE THE HORSE POWER REQUIRED TO ELEVATE A GIVEN QUANTITY OF WATER, TO A GIVEN HEIGHT, IN A GIVEN TIME.

RULE.—First, multiply the area of cross section of the piston or plunger of the pump, in square inches, by the height, in feet, required to elevate the water, and divide the product by the constant 2.304, *and the quotient will give the total weight of the column of water on the piston or plunger in pounds.*

Second, multiply the total weight of the column of water on the piston or plunger, in pounds, by the number of feet per minute the column of water is to move, and divide the product by 33000, *and the quotient will give the horse power required to elevate the water, without including friction.*

Example.—Let 16 inches equal diameter of the piston or plunger.
Let .7854 equal a constant.
Let 100 feet equal perpendicular height of column of water.
Let 2.304 equal a constant.
Let 150 feet equal speed of column of water per minute.
Let 33000 equal a constant.

Then we have:

$$\frac{[(16\times16\times.7854\times100)\div2.304]\times150}{33000}=39.66+ \text{ Horse power required.}$$

Performing the operation, we have:

$$\begin{array}{rl}
16 & \text{Diameter of piston or plunger.} \\
16 & \text{Diameter of piston or plunger.} \\
\hline
96 & \\
16 & \\
\hline
256 & \text{Square of diameter of piston or plunger.} \\
.7854 & \\
\hline
4\,7124 & \\
39\,270 & \\
157\,08 & \\
\hline
201.0624 & \text{Area of piston or plunger.}
\end{array}$$

Then, according to the rule, we multiply the area of the piston or plunger of the pump by the height of the column of water in feet, and divide the product by 2.304. Thus:

```
           201.0624  Area of piston or plunger.
                100  Height of column of water in feet.
          ----------
2.3040 ) 20106.2400 (8726+ lbs.  Total weight of water
         18432 0                 on piston or plunger.
         -------
          1674 24
          1612 80
          -------
            61 440
            46 080
            ------
            15 3600
            13 8240
            -------
```

Next, proceeding according to the rule, we multiply the total weight of column of water on piston or plunger by the speed per minute the column is required to travel, and then divide the product by 33000. Thus:

```
           8726  Total weight of column of water on piston
                 or plunger.
            150  Speed of column of water in feet per minute.
         -------
         436300
         8726
         -------
33000) 1308900 (39.66+  Horse power required.
        99000
        ------
        318900
        297000
        ------
         219000
         198000
         ------
          210000
          198000
          ------
```

NOTE.—A pound of water one inch square will make a column 2.304 feet high.

FEED PUMPS.

Section 4418 of the Revised Statutes of the United States, requires that in the equipment of steam vessels, inspectors shall see "that adequate and certain provision is made for an ample supply of water to feed the boilers at all times, whether such vessel is in motion or not."

The only way to provide an adequate supply of feed water without making the feed pumps unnecessarily large, is to regulate the size by the evaporative capacity of the boilers under the most favorable circumstances, which, in cylindrical two-flue marine boilers, has been

found to be seven pounds of water per pound of coal, and in cylindrical tubular land boilers ten pounds of water per pound of coal, as the average maximum evaporation in practice In the former the average maximum amount of coal consumed per square foot of grate surface is found to be twenty pounds, while in land boilers the consumption of coal varies from 12 to 40 pounds per square foot of grate surface. All of which proves conclusively that the capacity of the feed pump should be based upon the evaporative capacity of boilers. As Section 12 of Rule II of the Rules and Regulations of the United States Board of Supervising Inspectors, provides that the feed water for high pressure boilers shall in no case be less than 180° Fahrenheit, nor less than 100° Fahrenheit for low pressure boilers, it will be safe to assume an evaporation of seven pounds of water per pound of coal for marine boilers, and a consumption of 20 pounds of coal per square foot of grate surface per hour, which determines the capacity of feed pump required.

DIMENSIONS OF FEED PUMPS.

TO COMPUTE THE DIMENSIONS OF FEED PUMPS.

RULE.—First, multiply the number of pounds of coal consumed per square foot of grate surface per hour, by the number of pounds of water evaporated per pound of coal; then multiply the product by the number of boilers; then multiply that product by 27.64, *the number of cubic inches in one pound of water, and the answer will give the number of cubic inches required for evaporation; then multiply the number of cubic inches required for evaporation by* 1.12 *(which is adding twelve per cent. for leakage, etc.), and call the last product "Product No. 1."*

Second, multiply the number of displacement strokes of the feed pump by 60, *or if more than one pump, multiply the total number of displacement strokes of all the pumps by* 60; *then multiply the product by the length of stroke in inches; then multiply the last product by* .7854, *and call the resulting product "Product No. 2."*

Third, divide "Product No. 1" by "Product No. 2," and extract the square root of the quotient, and the answer will give the required diameter of the plunger or piston of feed pumps.

Example.—Let 20 pounds equal amount of coal consumed per square foot of grate surface per hour.
Let 7 pounds equal amount of water evaporated per pound of coal.
Let 25 square feet equal grate surface for each boiler.
Let 4 equal number of boilers.
Let 27.64 cubic inches of water equal one pound.

Let 1.12 equal a constant (used as a multiplier is equivalent to adding 12 per cent).
Let 60 equal a constant.
Let 2 equal number of single-acting pumps.
Let 20 equal number of displacement strokes for each pump per minute.
Let 12 inches equal length of stroke of each pump.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{20\times7\times25\times4\times27.64\times1.12}{60\times2\times20\times12\times.7854}}=4.37+ \text{ inches.}$$ Required diameter of plunger of each feed pump.

Performing the operation, we have:

20	Pounds of coal consumed per square foot of grate surface per hour.
7	Pounds of water evaporated per pound of coal.
140	
25	Square feet of grate surface for each boiler.
700	
280	
3500	
4	Number of boilers.
14000	Pounds of water evaporated per hour in four boilers.
27.64	Number of cubic inches in one pound of water.
560 00	
8400 0	
98000	
28000	
386960.00	Number of cubic inches of water evaporated per hour.
1.12	Multiplying by unity with per cent. annexed is equivalent to adding the per cent.
7739 2000	
38696 000	
386960 00	
433395.2000	"Product No. 1."

Performing the operation below the line in the above example, we have:

60	Number of minutes in one hour.
2	Number of pumps.
120	
20	Number of displacement strokes per minute for each pump.
2400	

Am't carried forward, 2400

Am't brought forward,

```
          2400
            12   Length of stroke of each pump.
          ----
          4800
         2400
         -----
         28800
         .7854   A constant.
         -----
        11 5200
       144 000
      2304 00
     20160 0
    ----------
    22619.5200   "Product No. 2."
```

Next, dividing "Product No. 1" by "Product No. 2," we have:

```
22619.52 ) 433395.20 ( 19.16+   Square of diameter of plunger.
           226195 2
           ---------
           207200 00
           203575 68
           ---------
             3624 320
             2261 952
             --------
             1362 3680
             1357 1712
             ---------
```

Finally, extracting the square root of the quotient, the square of the required diameter of the plunger, we have:

```
        19.16 ( 4.37+ inches.   Required diameter of plunger
        16                      of feed pumps.
        ---
  83 )  316
        249
        ----
 867 )  6700
        6069
        ----
```

That the student may have a better understanding of the subject, the rule relating to feed pumps will be given in detail, and in a more simplified form. And for the purpose of proving the correctness of the rule already given, and for convenient illustration, the dimensions used in the previous examples will be again taken.

RULE.—First, multiply the number of pounds of coal consumed per square foot of grate surface per hour by the number of pounds of water evaporated per pound of coal, and call the product "Product No. 1."

Second, multiply the number of square feet of grate surface in one boiler, by the number of similar boilers to be supplied with the required feed pump, and call the product "Product No. 2."

Third, multiply "Product No. 1," the number of pounds of water evaporated per one square foot of grate surface per hour, by "Product No. 2," the number of square feet of grate surface in the four boilers, and the product will give the total amount of water required for evaporation, and we will call this product "Product No. 3."

Fourth, as the pumps may not fill quite full during each stroke, and as there may be some leakage in pipes and boilers, and more or less water may pass off with the steam drawn from the boilers, we add 12 per cent. to the total amount of water required for evaporation; we then multiply "Product No. 3" by 1.12 and the product will give the total number of pounds of water required for evaporation with 12 per cent. added, and we will call this product "Product No. 4."

Fifth, we now reduce the number of pounds of water required to cubic inches by multiplying "Product No. 4" by 27.64, the number of cubic inches contained in one pound of water, and the product will give the total amount of water in cubic inches required per hour, and we will call the product "Product No. 5."

Sixth, divide "Product No. 5," the total amount of water in cubic inches required per hour by 60, and the quotient will give the total number of cubic inches of water required per minute, and we will call the quotient "Quotient No. 1."

Seventh, divide "Quotient No. 1," the number of cubic inches of water required per minute, by the number of displacement strokes per minute of all of the pumps combined (2 pumps, 20 displacement strokes each, equal 40), and the quotient will give the number of cubic inches of water required for each stroke, and this quotient we will call "Quotient No. 2."

Eighth, divide "Quotient No. 2," the number of cubic inches of water required for each stroke of the pump, by the length of the stroke in inches, and the quotient will give the number of cubic inches in the area of one inch in the length of the plunger, and call this quotient "Quotient No. 3."

Ninth, divide "Quotient No. 3," the area of one inch in the length of the plunger, by .7854, and the quotient will give the square of the required diameter of the plunger of each pump, and call the quotient "Quotient No. 4."

Tenth, extract the square root of "Quotient No. 4," the square of the diameter of pump plungers, and the answer will give the required diameter of each plunger.

Example.—First, multiplying the number of pounds of coal (20) consumed per square foot of grate surface per hour, by the number of pounds (7) of water evaporated per pound of coal, we have:

$$\begin{array}{r} 20 \\ 7 \\ \hline 140 \end{array}$$

"Product No. 1." Number of pounds of water evaporated per square foot of grate surface per hour.

Second, multiplying the number of square feet of grate surface in one boiler (25) by the number of boilers (4), we have:

$$\begin{array}{r} 25 \\ 4 \\ \hline 100 \end{array}$$

"Product No. 2." Square feet of grate surface in four boilers.

Third, multiplying "Product No. 1" by "Product No. 2," we have:

$$\begin{array}{r} 140 \\ 100 \\ \hline 14000 \end{array}$$

"Product No. 3." Pounds of water required per hour for evaporation.

Fourth, adding 12 per cent. to the amount of water required for evaporation, on account of leakage and pumps not filling quite full during each stroke, we multiply by unity with the per cent. annexed, which is equivalent to adding the per cent., and we have:

$$\begin{array}{r} 14000 \\ 1.12 \\ \hline 280\,00 \\ 1400\,0 \\ 14000 \\ \hline 15680.00 \end{array}$$

"Product No. 4." Total number of pounds of feed water required per hour.

Fifth, reducing the number of pounds of feed water required to cubic inches by multiplying "Product No. 4" by 27.64, the number of cubic inches in a pound of water, we have:

$$\begin{array}{r} 15680 \\ 27.64 \\ \hline 627\,20 \\ 9408\,0 \\ 109760 \\ 31360 \\ \hline 433395.20 \end{array}$$

"Product No. 5." Total number of cubic inches of water required per hour.

Sixth, dividing "Product No. 5," the total number of cubic inches of water required per hour, by 60, we have:

```
60 ) 433395.20 ( 7223.25+     "Quotient No. 1." Number of
     420                         cubic inches of water re-
     ---                         quired per minute.
      133
      120
      ---
       139
       120
       ---
        195
        180
        ---
         152
         120
         ---
          320
          300
          ---
```

Seventh, dividing "Quotient No. 1," the number of cubic inches of water required per minute, by 40 (20 for each pump), the number of displacement strokes of the pumps per minute, and we have:

```
40 ) 7223.25 ( 180.58+     "Quotient No. 2." Number of cu-
     40                       bic inches of water required for
     ---                      each stroke of the pumps.
     322
     320
     ---
       232
       200
       ---
        325
        320
        ---
```

Eighth, dividing "Quotient No. 2," the number of cubic inches of water required for each displacement stroke of the pumps, by 12 inches (the length of the stroke), we have:

```
12 ) 180.58 ( 15.0483+     "Quotient No. 3." Number of
     12                       cubic inches in the area of one
     --                       inch in the length of plunger.
     60
     60
     ---
       58
       48
       ---
       100
        96
        ---
         40
         36
         --
```

Ninth, dividing "Quotient No. 3," the number of cubic inches in the area of one inch in the length of the plunger, by .7854, we have:

```
.7854 ) 15.0483 ( 19.16+    "Quotient No. 4." The square
         7 854                of the required diameter of
         ------               plunger.
         7 1943
         7 0686
         -------
           12570
            7854
           ------
            47160
            47124
            -----
```

Tenth, extracting the square root of "Quotient No. 4," the square of the required diameter of the plunger, we have:

```
       19.16 ( 4.37+ inches.  Required diameter of plunger.
       16
       ----
 83 )  316
       249
       -----
867 )  6700
       6069
       -----
```

CHAPTER IX.

SMOKE STACK PROPORTIONS.

AREA OF STACK.

TO DETERMINE THE AREA OF REQUIRED STACK.

RULE.—Multiply the area of cross section of tubes or flues in the boiler by the constant 1.3, *and the product will give the required area.*

Example.—Let 4 inches equal diameter of tubes.
Let .7854 equal a constant.
Let 40 equal number of tubes.
Let 1.3 equal a constant.

Then we have: $4 \times 4 \times .7854 \times 40 \times 1.3 = 653.4528$ square inches. Required area of stack.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r}
4 \\
4 \\
\hline
16 \\
.7854 \\
\hline
4\,7124 \\
7\,854 \\
\hline
12.5664 \\
40 \\
\hline
502.6560 \\
1.3 \\
\hline
150\,79680 \\
502\,6560 \\
\hline
653.45280
\end{array}$$

653.45280 square inches. Required area of stack.

DIAMETER OF SQUARE STACK.

TO DETERMINE THE DIAMETER OF A SQUARE STACK.

RULE.—Extract the square root of the required area in square inches of stack, and the answer will give the diameter of a square stack.

Example.—Let 653.4528 square inches equal required area of stack.

Then we have: $\sqrt{653.4528} = 25.56+$ inches. Required diameter of a square stack.

Performing the operation, we have:

```
      653.4528(25.56+ inches.   Required diameter of
      4                         square stack.
45)   253
      225
505)  2845
      2525
5106) 32028
      30636
```

DIAMETER OF ROUND STACK.

TO DETERMINE THE DIAMETER OF A ROUND STACK.

RULE.—Divide the required area of the stack by .7854 *and extract the square root of the quotient, the answer will give the diameter of a round stack.*

Example.—Let 653.4528 square inches equal required area of stack.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{653.4528}{.7854}} = 28.84+ \text{ inches.}$$ Required diameter of round stack

Performing the operation, we have:

```
.7854) 653.4528 (832   Square of required diameter.
       628 32
        25 132
        23 562
         1 5708
         1 5708
```

Then, extracting the square root of the quotient (the square root of the required diameter), we have:

```
      832 (28.84+ inches.   Required diameter of
      4                     round stack.
48)   432
      384
568)  4800
      4544
5764) 25600
      23056
```

PERCENTAGE OF AIR SPACE FOR GRATE BARS.

The required height of the stack depends, in a very large measure, upon conditions:

First, no grate bars should be allowed in any boiler furnace burning bituminous coal, with less than 60 per cent. of air space. All calculations based upon grate surface can not but be erroneous unless the air space is taken into account. Therefore, grate bars with contracted air space require a higher stack than grate bars with ample air space.

Second, much depends upon the fuel used; wood requires less force of draft than lump coal, and lump coal less than nut, and nut less than slack.

When we speak of 60 per cent. of air space, the student must understand that the air space is not governed by the width of the face of the bars, but the thickness of metal in the bars must be governed by the width of the air space required; therefore, if ⅝ of an inch be required for air space, the thickness of metal in the bars must not be more than 42 one hundredths of an inch.

PROPORTION AND ARRANGEMENT OF BREECHING.

The breeching should contain an area of cross section of 15 per cent. larger than the area of cross section of the tubes or flues, but no larger, and in no case should the area be less than the area of cross section of the flues. In the construction of the breeching care should in all cases be taken to prevent the current of the gases of one boiler cutting across the current of the gases from any other boiler in a battery. The uptake for each boiler should be so constructed that the gases from the boilers will not come in contact with each other until they have been turned in the same direction toward the stack, and all sharp angles should be avoided.

FORCE OF DRAUGHT REQUIRED.

As the height of the stack should be governed by the force of draught required, and as practice has demonstrated that ½ inch of water, as shown by a water-gauge, is required for bituminous lump coal to burn it to advantage, and ¾ of an inch for nut, not less than 1⅛ of an inch should be provided for slack.

HEIGHT OF STACK—BITUMINOUS LUMP COAL.

TO DETERMINE THE HEIGHT OF STACK FOR BITUMINOUS LUMP COAL.

RULE.—Divide the draught required in inches of water by the constant .0073, *and the quotient will give the required height of the stack in feet.*

Example.—Let 5 tenths of an inch of water equal force of required draught for lump coal.

Let .0073 equal a constant.

Then we have:

```
.0073 ).5000 ( 68.5— feet.   Required height of stack for
         438                   bituminous lump coal.
         ———
          620
          584
          ———
           360
           365
           ———
```

HEIGHT OF STACK—BITUMINOUS NUT COAL.

TO DETERMINE THE HEIGHT OF STACK FOR BITUMINOUS NUT COAL.

Example.—Let 75 one hundredths of an inch of water equal force of required draught for nut coal.

Let .0073 equal a constant.

Then we have:

```
.0073 ) .7500 ( 102.74— feet.   Required height of stack for
         73                       bituminous nut coal.
         ———
          200
          146
          ———
           540
           511
           ———
            290
            292
            ———
```

HEIGHT OF STACK—BITUMINOUS SLACK COAL.

TO DETERMINE THE HEIGHT OF STACK FOR BITUMINOUS SLACK COAL.

Example.—Let 1.125 inches of water equal force of required draught for slack coal.

Let .0073 equal a constant.

Then we have:

```
.0073 ) 1.1250 ( 154+ feet.   Required height of stack for
         73                     bituminous slack coal.
         ———
          395
          365
          ———
           300
           292
           ———
```

CHAPTER X.

SPECIFICATIONS AND PLANS FOR TUBULAR BOILERS.

SPECIFICATIONS FOR BOILERS.

There will be two horizontal return tubular boilers, each 16 feet in length from outside to outside of heads, and sixty inches in diameter, measured on the outside of the smallest ring, the first ring to extend 15 inches beyond the front head to form a smoke box. The boilers to be set in a battery with a dividing wall between them and so connected that each may be operated independent of the other, or that both may be operated together.

MATERIAL, SHELL AND HEADS.

The shell plates are to be made of open hearth homogeneous flange steel, $\frac{5}{16}$ of an inch in thickness, and have a tensile strength of not less than 60,000 pounds, nor more than 65,000 pounds, per square inch of section. A test piece shall be cut from each plate, and each test piece shall show an elongation in testing of not less than 25 per cent. in 8 inches, or a reduction of area at point of fracture of not less than 50 per cent. The heads will be made of open hearth homogeneous flange steel, $\frac{5}{8}$ of an inch in thickness.

TEST PIECES.

The test section of one-half of the test pieces shall be eight inches long and $\frac{8}{10}$ of an inch wide for determining elongation and tensile strength, and one-half of the test pieces shall be made according to the rules of the United States Board of Supervising Inspectors of Steam Vessels, for determining reduction of area.

RIVETS AND RIVET HOLES.

All rivets will be made of the best quality of steel. All longitudinal seams will be double-riveted zig-zag, and placed above the fire line, the lower sheets will lap over the outer side of the upper sheets. The rivet holes for longitudinal seams are to be $\frac{11}{16}$ inch in diameter, and to have a pitch of $2\frac{1}{2}$ inches from center to center of rivet holes; and a distance of $1\frac{1}{4}$ inches from the center of the outer row of rivet holes to the center of the inner row of rivet holes. The center of outer row of rivet holes will not be less than $1\frac{3}{32}$ inches from the edge

of the sheet. All edges of sheets will be planed true and neatly and efficiently calked. All rivet holes for boilers will be fairly drilled, so that they will come fair without the use of a drift pin forcibly, and without the use of a reamer if such use will enlarge any rivet hole materially beyond the diameter herein specified. The circular seams will be single riveted, and the rivet holes for such seams will be $\frac{3}{4}$ inch in diameter, and the center of rivet holes will be placed not less than $1\frac{3}{32}$ inches from the edge of the sheet and have a pitch of $2\frac{1}{4}$ inches from center to center of rivet holes. In no case will any shoulder or off-set be allowed in any rivet hole; and in all cases must each rivet be driven so as to fill the rivet hole completely.

BRACING.

There will be six braces each $1\frac{1}{4}$ inches in diameter in the front head above the tubes, and attached to the head by heavy tee or angle-iron, and in such a manner as to relieve the tubes of any undue strain from the steam or hydrastatic pressure; and they will be attached to the shell by three $\frac{3}{4}$ inch rivets. The back head will have five such braces attached in a similar manner above the tubes; except that the braces on each side of the man hole will be attached to the head by means of crow feet, with two $\frac{7}{8}$ inch rivets; and all braces attached to the heads shall not be less than $3\frac{1}{2}$ feet in length. There will be two rods of $1\frac{1}{4}$ inch round iron, running from head to head below the tubes, and screwed tightly into each head and securely riveted.

TUBES.

The tubes are to be the standard American boiler tubes or their equivalent; and each boiler to contain 52 tubes 4 inches in diameter and 16 feet in length, and set in straight rows vertical and horizontal, with a clear space between the tubes of not less than 1 inch, and between the tubes and shell of the boiler a space of not less than 3 inches, and the tubes shall have a central vertical space of not less than $2\frac{1}{2}$ inches.

MAN HOLES.

There will be two man holes in each boiler, one in the back head above the tubes and one in the front head below the tubes. Each such man hole shall have an opening of not less than 11 x 15 inches; and each such man hole shall be flanged inwardly to a depth of not less than $1\frac{1}{2}$ inches, and the faces of such flanges shall be planed or otherwise brought to a true surface, and each such man hole shall be provided with the Eclipse man head (as shown in Figs. 72 and 73 of the plans).

WALL BRACKETS.

There will be two cast-iron brackets securely riveted on each side of the boilers on a line with the upper row of tubes. Each bracket to be of sufficient size and weight to carry the boilers with perfect safety, and to have a width of not less than 9 inches, and project from the shell of the boiler a distance of not less than 12 inches. The forward brackets to be provided with a cast-iron bearing plate not less than 10 inches wide, 18 inches long and 2 inches thick. The back brackets to be provided each with similar plates and two wrought-iron rollers, each 10 inches in length and 1 inch in diameter.

DOME.

There will be a dome 30 inches in diameter and 30 inches in height, made of the same quality of material as that in the shell of the boiler. And the dome shall have a thickness of material of $\frac{5}{16}$ of an inch, and shall be properly flanged and double riveted to the shell of the boiler. The dome head shall be of open hearth flange steel, and have a thickness of material of $\frac{3}{8}$ of an inch, and to be bumped to a radius of not more than 20 inches.

SMOKE BOX.

The smoke box will be fitted with a collar riveted to the top to form a base for connecting the breeching.

FIRE FRONT AND CASTINGS.

The boilers will each be provided with a full, flush fire-front of neat design, fire doors, ash pit doors, a cleaning door 12 x 16 inches for back end, liners, grate bearers, anchor bolts, six buck staves and tee rods for the same, a full set of grate bars 4 feet in length, and having not less than 60 per cent. of air space

FITTINGS.

The boilers will each be provided with one 3 inch spring-loaded safety-valve, made according to the rules of the United States Board of Supervising Inspectors, one 10 inch steam-gauge, one $2\frac{1}{2}$ inch asbestos blow-off valve, one 2 inch check-valve, one 14 inch water-gauge with $\frac{5}{8}$ x 12 inch glass, three $\frac{3}{4}$ inch gauge-cocks, all fitted to a Williams' safety water column (as shown in Figs. 74 and 75), or its equivalent, one set of fire irons, consisting of one hoe, one slice bar and one scraper.

INSPECTION.

Each boiler when completed, is to be subjected to a hydrastatic pressure of 185 pounds per square inch, in the presence of the inspector of any reliable boiler insurance company, and the certificate of inspection of such company, and policy of insurance for five hundred dollars

to be furnished the purchaser of the boiler. All of the material and workmanship about the boilers to be first-class, and to be open to inspection, at all stages of construction, by any representative of the purchaser.

PLANS OF BOILERS.

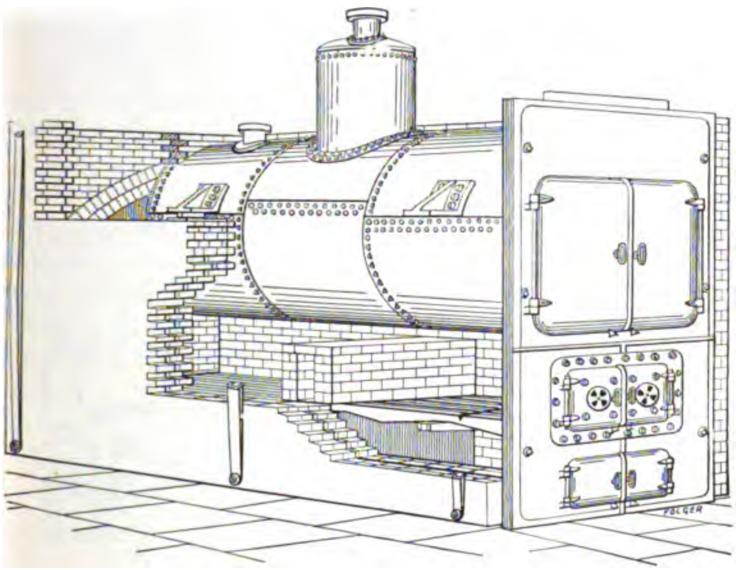


Fig. 70.
HORIZONTAL TUBULAR BOILER—FRONT VIEW.

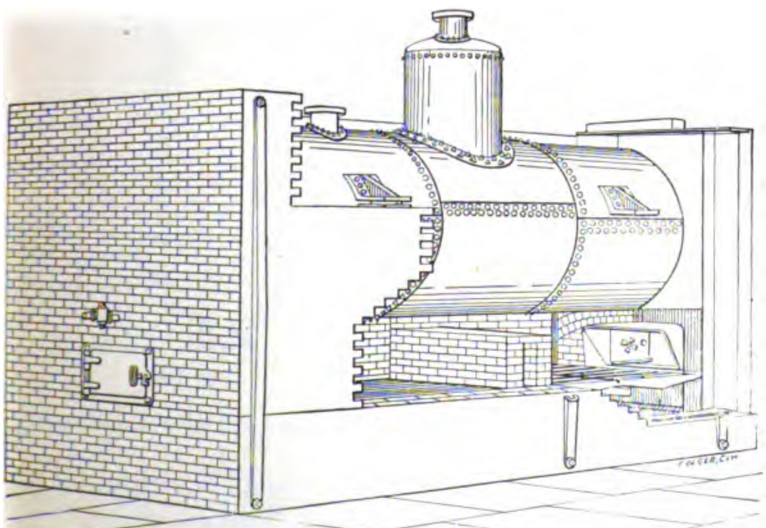


Fig. 71.
HORIZONTAL TUBULAR BOILER—BACK VIEW.

THE ECLIPSE MAN HOLE AND COVER.

Fig. 72.

REAR HEAD OF TUBULAR BOILER.

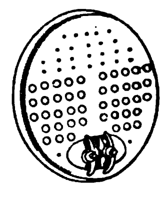


Fig. 73.

FRONT HEAD OF TUBULAR BOILER.

THE PITTSBURGH SAFETY WATER COLUMN.

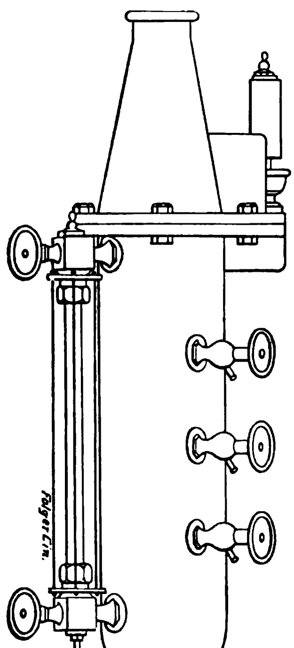


Fig. 74.

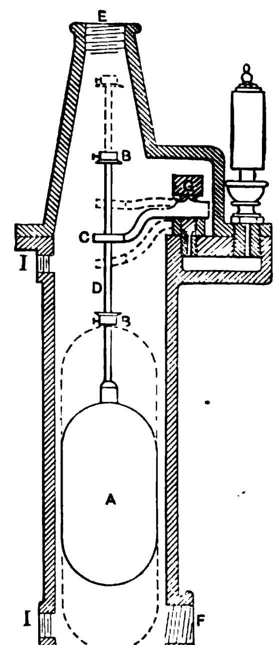


Fig. 75.

Figs. 74 and 75 show the general arrangement of the safety water column.

Fig. 74 represents the column trimmed complete with water-gauge and gauge-cocks, ready for connection to the boiler.

Fig. 75 shows detailed sectional drawing of the column with working parts. The steam connection is made at the top E, and the water connection at the lower end at F, or through the blow-off H, at the bottom. The column is placed on the boiler, so that the center of the lower gauge-cock is 2 or 2½ inches above the tube or flue level.

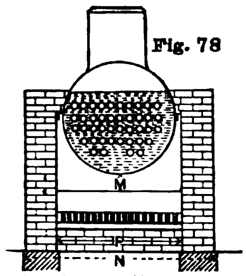
When the water falls in the boiler, it allows the seamless copper float A to drop to the lower gauge-cock, when the upper knocker B, on the float-rod D, strikes the lever C, opens the valve G, and allows the steam to escape through the whistle, and sound the low water alarm. In the same manner, when the water raises the float to the center of the upper gauge-cock, the lower knocker B raises the lever C, sounding the whistle for high water.

I I, Fig. 75, is where the water-gauge is attached, and that gauge indicates the exact height of the water in the boiler. The blow-off connection is made at H. The dotted lines representing the lever C in two different positions, represent the points at which the alarms are given for high and low water.

SINGLE SETTING, WITH FULL FRONT.

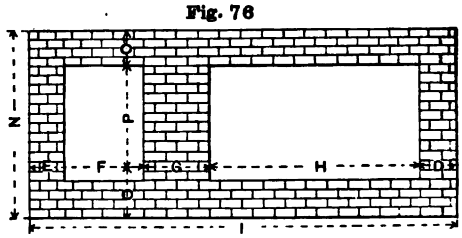
Plan A.

Fig. 78



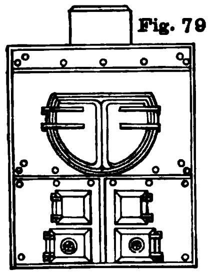
Section on line A L K.

Fig. 76



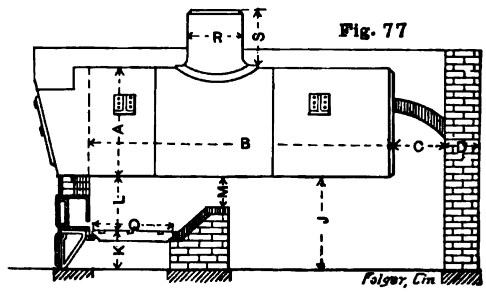
Ground Plan.

Fig. 79



Front Elevation.

Fig. 77

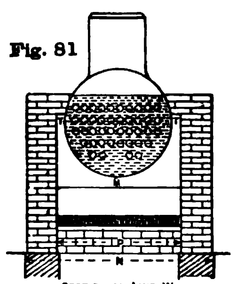


Longitudinal Sectional Elevation.

SINGLE SETTING, WITH HALF ARCH FRONT.

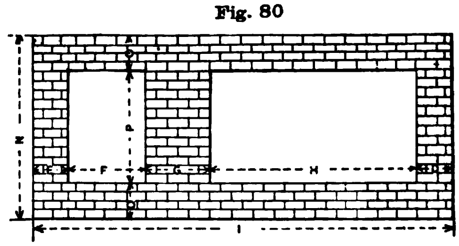
Plan B.

Fig. 81



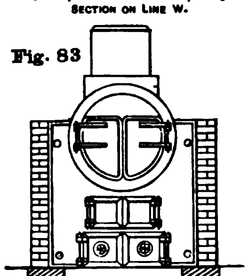
Section on Line W.

Fig. 80



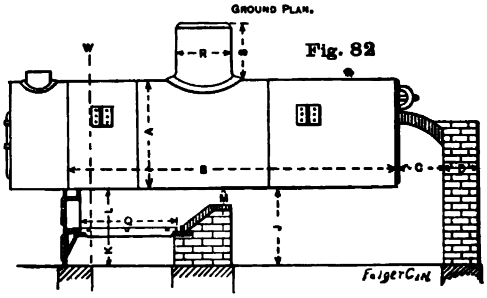
Ground Plan.

Fig. 83



Front Elevation.

Fig. 82



Longitudinal Sectional Elevation.

DIMENSIONS FOR SINGLE SETTING.—PLAN A.

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	No. of Fire Brick.	No. of Common Brick.
In.	Ft.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Above Floor Level.	
30	8	16	13	9	34	13	65	134	36	18	18	4	64	13	38	36	18	20	4	650	4000
36	8	16	13	9	34	13	65	134	36	18	18	4	70	13	44	36	20	22	4	800	4500
36	10	18	13	9	40	13	85	160	36	18	18	4	70	13	44	42	20	22	4	950	4900
40	10	18	13	9	40	13	85	160	44	24	20	6	74	13	48	42	22	24	4	1020	5600
42	10	18	13	9	40	13	85	160	44	24	20	6	76	13	50	42	22	24	4	1080	5800
44	10	18	13	13	40	18	80	164	44	24	20	6	78	13	52	42	24	24	4	1150	5900
44	12	18	13	13	46	18	98	188	44	24	20	6	78	13	52	48	24	24	4	1300	6200
48	12	20	18	18	46	18	102	202	44	24	20	8	96	18	60	48	28	28	6	1500	8100
48	14	20	18	18	52	18	120	226	44	24	20	8	96	18	60	54	28	28	6	1650	9000
54	12	20	18	18	46	18	102	202	48	24	24	8	102	18	66	48	30	30	6	1650	9000
54	14	20	18	18	52	18	120	226	48	24	24	8	102	18	66	54	30	30	6	1850	9900
60	12	24	22	18	46	22	100	208	48	24	24	8	116	22	72	48	34	34	6	1900	12000
60	14	24	22	18	52	22	118	232	48	24	24	8	116	22	72	54	34	34	6	2075	12600
66	12	24	22	18	46	22	100	208	48	24	24	8	122	22	78	48	36	36	6	2075	12500
66	14	24	22	18	52	22	118	232	48	24	24	8	122	22	78	54	36	36	6	2275	13200
66	16	24	22	18	58	22	136	256	48	24	24	8	122	22	78	60	36	36	6	2475	13900

DIMENSIONS FOR SINGLE SETTING.—PLAN B.

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	No. of Fire Brick.	No. of Common Brick.
In.	Ft.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Above Floor Level.	
30	8	16	13	9	34	13	60	129	36	18	18	4	64	13	38	36	18	20	4	600	3250
36	8	16	13	9	34	13	60	129	36	18	18	4	70	13	44	36	20	22	4	760	3550
36	10	18	13	9	40	13	80	155	36	18	18	4	70	13	44	42	20	22	4	900	4100
40	10	18	13	9	40	13	80	155	44	24	20	6	74	13	48	42	22	24	4	980	4350
42	10	18	13	9	40	13	80	155	44	24	20	6	76	13	50	42	22	24	4	1040	4500
44	10	18	13	13	40	18	76	160	44	24	20	6	78	13	52	42	24	24	4	1100	4650
44	12	18	13	13	46	18	94	184	44	24	20	6	78	13	52	48	24	24	4	1250	5300
48	12	20	18	18	46	18	93	193	44	24	20	8	96	18	60	48	28	28	6	1450	7100
48	14	20	18	18	52	18	111	217	44	24	20	8	96	18	60	54	28	28	6	1600	7900
54	12	20	18	18	46	18	93	193	48	24	24	8	102	18	66	48	30	30	6	1600	8100
54	14	20	18	18	52	18	111	217	48	24	24	8	102	18	66	54	30	30	6	1800	9000
60	12	24	22	18	46	22	91	199	48	24	24	8	116	22	72	48	34	34	6	1850	11000
60	14	24	22	18	52	22	109	223	48	24	24	8	116	22	72	54	34	34	6	2000	11900
66	12	24	22	18	46	22	91	199	48	24	24	8	122	22	78	48	36	36	6	2000	11700
66	14	24	22	18	52	22	109	223	48	24	24	8	122	22	78	54	36	36	6	2200	12300
66	16	24	22	18	58	22	127	247	48	24	24	8	122	22	78	60	36	36	6	2400	12900

SINGLE SETTING, WITH FULL FRONT.—PLAN C.

Fig. 85

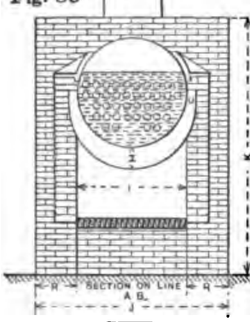
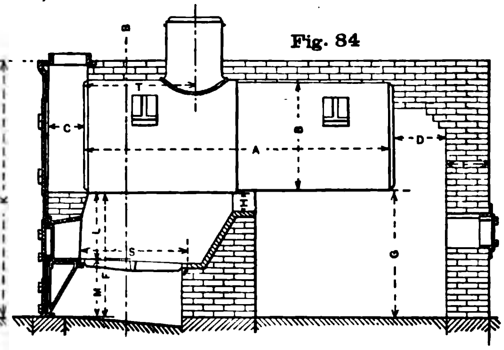
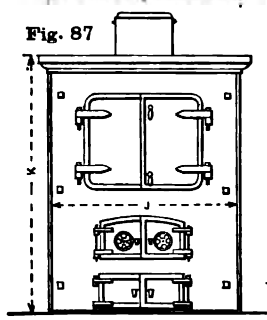


Fig. 84



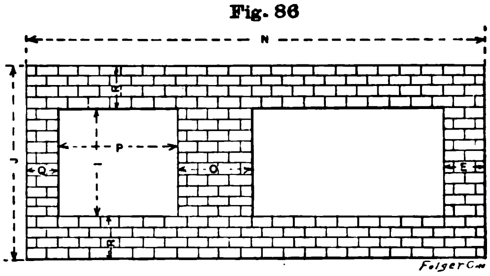
LONGITUDINAL SECTIONAL ELEVATION.

Fig. 87



FRONT ELEVATION.

Fig. 86



GROUND PLAN.

SINGLE SETTING, WITH HALF ARCH FRONT.—PLAN D.

Fig. 89

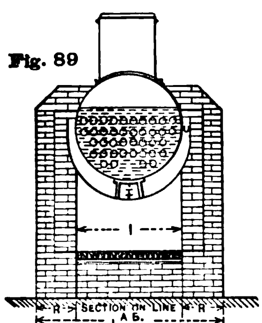
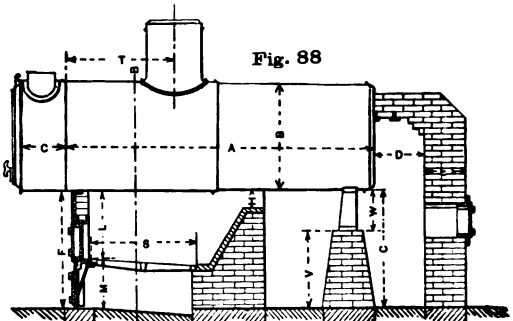
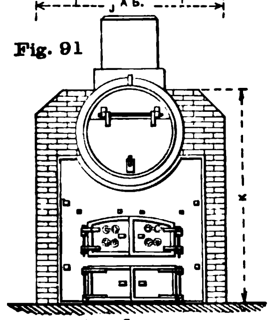


Fig. 88



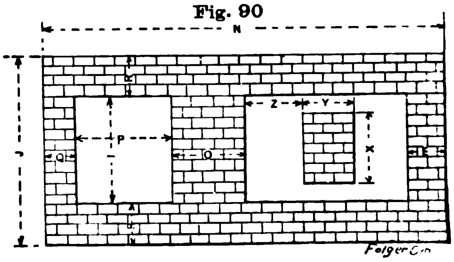
LONGITUDINAL SECTIONAL ELEVATION.

Fig. 91



FRONT ELEVATION.

Fig. 90



GROUND PLAN.

DIMENSIONS FOR SINGLE SETTING.—PLAN C.

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	No. of Fire Brick	No. of Common Brick.
Ft.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Ft. In	In.	In.	In.	In.	In.	In.	In.	Above Floor Level.	
7	32	12	20	16	45	44	7	32	64	85	26	19	11 6	20	40	12	16	36	34	4	600	6800
7	34	12	20	16	48	47	8	34	66	90	26	22	11 6	20	40	12	16	36	34	4	600	7500
8	36	12	20	16	48	47	8	36	68	92	26	22	12 6	24	40	12	16	36	34	4	650	7700
10	36	12	20	16	48	47	8	36	68	92	26	22	14 6	28	46	12	16	42	42	4	720	8500
8	42	14	20	16	48	47	8	42	74	98	27	21	12 8	24	40	12	16	36	34	4	730	8500
10	42	14	20	16	48	47	8	42	74	98	27	21	14 8	28	46	12	16	42	42	4	770	9600
10	44	14	24	16	48	47	10	44	76	100	27	21	15 0	28	46	12	16	42	42	4	880	10500
12	44	14	24	16	48	46½	10	44	76	100	27	21	17 0	32	52	12	16	48	49	4	940	10800
14	44	14	24	16	48	46½	10	44	76	100	27	21	19 0	36	58	12	16	54	84	4	1120	11500
12	48	16	24	16	47	45½	10	48	88	103	26	21	17 2	32	52	12	20	48	49	4	1120	13600
14	48	16	24	16	47	45½	10	48	88	103	26	21	19 2	36	58	12	20	54	84	4	1140	15700
12	54	16	24	20	50	48½	10	54	94	112	26	24	17 6	32	52	12	20	48	49	4	1160	16200
15	54	16	24	20	50	48½	10	54	91	112	26	24	20 8	36	56	16	20	54	90	4	1270	17500
12	60	18	24	20	50	48½	12	60	108	118	26	24	17 10	32	50	16	24	48	49	4	1400	20500
14	60	18	24	20	50	48½	12	60	108	118	26	24	19 10	36	56	16	24	54	84	4	1500	23000
16	60	18	26	20	50	48	12	60	108	118	26	24	22 0	40	56	16	24	54	96	4	1540	25300
15	66	18	28	20	50	48½	12	66	114	124	26	24	21 2	36	56	16	24	54	90	4	1590	26000
16	66	18	28	20	50	48	12	66	114	124	26	24	22 2	40	56	16	24	54	96	4	1620	27000
16	72	20	30	20	50	48	12	72	120	130	26	24	22 6	40	56	16	24	54	96	4	1750	30000

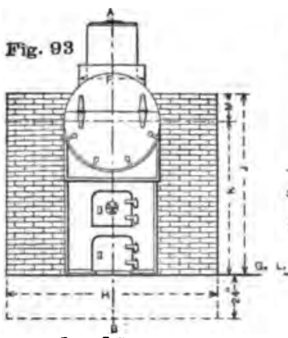
DIMENSIONS FOR SINGLE SETTING.—PLAN D.

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	No. of Fire Brick.	No. of Common Brick.
Ft.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Ft. In	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	Above Floor Level.	
7	32	14	20	16	46	45	7	32	64	73	26	20	10 3	20	33	12	16	36	34	4	36	9	24	12	7	600	6150
7	34	14	20	16	46	45	8	34	66	75	26	20	10 3	20	33	12	16	36	34	4	36	9	28	12	7	600	6200
8	36	14	20	16	46	45	8	36	68	77	26	20	11 3	24	33	12	16	36	34	4	36	9	28	12	15	650	6700
10	36	14	20	16	46	45	8	36	68	77	26	20	13 3	28	39	12	16	42	42	4	36	9	28	12	25	720	7050
8	42	18	20	16	46	45	8	42	74	83	27	19	11 3	24	32½	12	16	36	34	4	32¾	12¼	32	16	11	730	7700
10	42	18	20	16	46	45	8	42	74	83	27	19	13 3	28	38½	12	16	42	42	4	32¾	12¼	32	16	25	770	8700
10	44	18	24	16	46	45	10	44	76	85	27	19	13 7	28	38½	12	16	42	42	4	32¾	12¼	36	16	25	880	8800
12	44	18	24	16	46	44½	10	44	76	85	27	19	15 7	32	44½	12	16	48	49	4	32¼	12¼	36	20	35	940	9300
14	44	18	24	16	46	44½	10	44	76	85	27	19	17 7	36	50½	12	16	54	84	4	32¼	12¼	36	24	45	1120	9500
12	48	19	24	16	50	48½	10	48	88	93	26	24	15 7	32	48	12	20	48	49	4	36¼	12¼	36	20	35	1120	11100
14	48	19	24	16	50	48½	10	48	88	93	26	24	17 7	36	54	12	20	54	84	4	36¼	12¼	36	24	45	1140	12900
12	54	19	24	20	50	48½	10	54	94	99	26	24	15 11	32	48½	12	20	48	49	4	34	14½	42	20	35	1160	13200
15	54	19	24	20	50	48½	10	54	94	99	26	24	19 1	36	52½	16	20	54	90	4	34	14½	42	24	57	1270	14200
12	60	21	24	20	46¾	45½	12	60	108	101¾	26	20¾	16 1	32	47	16	24	48	49	4	31	14½	48	20	37	1400	16700
14	60	21	24	20	46¾	45	12	60	108	101¾	26	20¾	18 1	36	53	16	24	54	84	4	30½	14½	48	24	45	1500	19200
16	60	21	26	20	46¾	45	12	60	108	101¾	26	20¾	20 3	40	53	16	24	54	96	4	30½	14½	48	24	65	1540	21500
15	66	24	28	20	47	45½	12	66	114	108	26	21	19 5	36	52½	16	24	54	90	4	31	14½	54	24	57	1590	22100
16	66	24	28	20	47	45½	12	66	114	108	26	21	20 5	40	52½	16	24	54	96	4	31	14½	54	24	65	1620	23100
16	72	24	30	20	48	46½	12	72	120	115	28¼	19¾	20 7	40	52½	16	24	54	96	4	27½	19	54	24	65	1750	26000

SINGLE SETTING, WITH HALF ARCH FRONT AND BOILER RESTING ON FIRE FRONT AND BACK STAND.

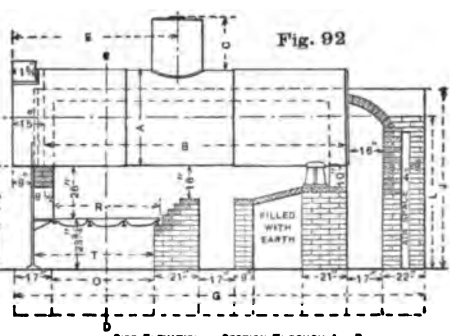
PLAN E.

Fig. 93



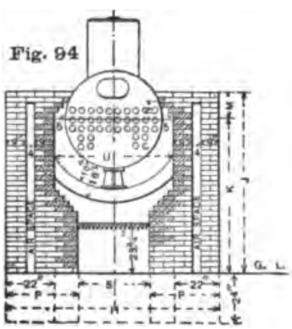
FRONT ELEVATION.

Fig. 92



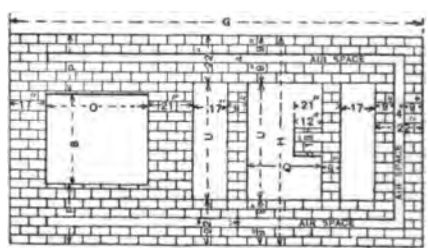
SIDE ELEVATION SECTION THROUGH A. B.

Fig. 94



SECTION THROUGH G. D.

Fig. 95



GROUND PLAN.

SPECIFICATIONS FOR SINGLE SETTING.—PLAN E.

Nom'l Rated Horse Power.	SHELL. Diam In.	SHELL. Lgth Ft.	Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE. Shell In.	THICKNESS OF BOILER PLATE. Heads In.	BOILER PLATE Homogeneous Steel. Tensile Strength per Square Inch.	Safe Working Pressure for Boiler per Square Inch. Longitudinal Seams, Double Rivet'd.	DOME. Diam In.	DOME. Hght In.	Size of Steam Outlet, Inches.	TUBES. No.	TUBES. Diam In.	TUBES. Lgth Ft.	Total Heating Surface, Square Feet.	STACK. Diam In.	STACK. Lgth Ft.	SHIPPING WEIGHT. Pounds Approximate. Boiler Only.	Fixtures Only.	Boiler and Fixtures.
25	36	12	15	1/4	3/8	60,000 lbs.	166 lbs.	18	20	2½	26	3	12	318	18	35	2,970	1,830	4,800
30	40	12	15	1/4	3/8	60,000 "	150 "	20	22	2½	34	3	12	404	20	40	3,441	2,159	5,600
35	42	12	15	1/4	7/16	60,000 "	143 "	20	22	2½	40	3	12	464	22	40	3,760	2,240	6,000
40	46	12	15	1/4	7/16	60,000 "	130 "	24	24	3	42	3	12	491	22	40	4,345	2,355	6,700
45	48	12	15	9/32	7/16	60,000 "	140 "	24	24	3	48	3	12	551	24	40	5,045	2,355	7,400
55	52	14	15	9/32	7/16	60,000 "	130 "	26	26	3½	44	3½	14	693	26	40	6,620	2,880	9,500
60	54	14	15	5/16	1/2	60,000 "	139 "	26	26	3½	46	3½	14	721	26	40	7,300	3,000	10,300
70	54	16	15	5/16	1/2	60,000 "	139 "	26	26	3½	40	4	16	817	26	40	8,711	3,089	11,800
75	60	14	15	5/16	1/2	60,000 "	126 "	28	28	4	62	3½	14	940	30	40	9,180	3,420	12,600
85	60	16	15	5/16	1/2	60,000 "	126 "	28	28	4½	52	4	16	1,045	30	40	10,643	3,457	14,100
100	66	16	15	3/8	1/2	60,000 "	136 "	30	30	4½	64	4	16	1,265	36	40	13,294	4,206	17,500
125	72	16	15	3/8	1/2	60,000 "	125 "	30	30	5	82	4	16	1,578	40	40	15,900	4,700	20,600

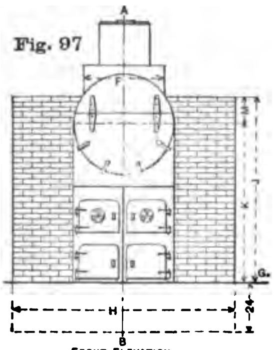
DIMENSIONS FOR SINGLE SETTING.—PLAN E.

Nominal Rated Horse Power of Boiler.	BOILERS. Diameter of Boiler. A IN.	Length of Boiler. B FT.	Top of Boiler to Top of Dome Flange. C IN.	Center of Independent Dome to Front Boiler. D FT. IN.	Center of Riveted Dome to Front of Boiler. E FT. IN.	Width of Stack Saddle. F FT. IN.	SETTING. Length of Foundation. G FT. IN.	Width of Foundation. H FT. IN.	Floor Line to Top of Safety Valve. I FT. IN.	Height of Walls. J FT. IN.	Floor Line to Center of Boiler, Front End. K FT. IN.	Floor Line to Center of Boiler, Back End. L FT. IN.	Center of Boiler to Top of Walls. M IN.	Center of Boiler to Closing in of Side Walls. N IN.	Front Wall to Bottom of Bridge Wall. O FT. IN.	Thickness of Jamb. P FT. IN.	Length of Apron. Q FT. IN.	FURNACE. Length of Furnace. R FT. IN.	Width of Furnace. S FT. IN.	Length of Ash Pit. T FT. IN.	Width of Soot and Back Pits. U FT. IN.	Number of Sq. Feet in Grate Surface. SQ. FT.
25	36	12	21	7 2	6 6	1 8	16 4	7 6	10 4	6 7	5 8	5 7	11	6	2 9	2 1	4 3	2 10	3 4	3 6	3 10	9.44
30	40	12	23	7 2	6 6	2 2	16 4	7 10	10 10	6 10	5 10	5 9	12	7	3 3	2 3	3 9	3 4	3 4	4 0	4 2	11.11
35	42	12	23	7 2	6 6	2 8	16 4	8 0	11 0	6 11	5 11	5 10	12	7	3 7	2 3	3 5	3 8	3 6	4 4	4 4	12.83
40	46	12	25	7 2	6 6	2 8	16 4	8 4	11 7	7 1	6 1	6 0	12	7	4 1	2 5	2 11	4 2	3 6	4 10	4 8	14.58
45	48	12	25	7 2	6 6	3 6	16 4	8 6	11 9	7 2	6 2	6 1	12	7	4 1	2 5	2 11	4 2	3 8	4 10	4 10	15.28
55	52	14	27	8 7	7 6	3 9	18 4	9 0	12 3	7 6	6 4	6 3	14	9	4 1	2 6	4 11	4 2	4 0	4 10	5 4	16.66
60	54	14	27	8 7	7 6	3 9	18 4	9 2	12 5	7 6	6 5	6 4	13	8	4 1	2 5	4 11	4 2	4 4	4 10	5 6	18.05
70	54	16	27	9 10½	8 6	3 9	20 4	9 2	12 5	7 6	6 5	6 4	13	8	4 1	2 1	6 11	4 2	5 0	4 10	5 6	20.83
75	60	14	33	8 7	7 6	4 7	18 4	9 8	13 6	7 9	6 8	6 7	13	8	4 1	2 3	4 11	4 2	5 2	4 10	6 0	21.52
85	60	16	33	9 10½	8 6	4 7	20 4	9 8	13 6	8 1	6 8	6 7	17	12	4 1	2 2	6 11	4 2	5 4	4 10	6 0	22.22
100	66	16	33	9 10½	8 6	5 0	20 4	10 2	14 4	8 5	6 11	6 10	18	13	4 7	2 5	6 5	4 8	5 4	5 4	6 6	24.88
125	72	16	33	9 10½	8 6	5 6	20 4	10 8	14 7	8 6	7 2	7 1	16	11	4 10	2 4	6 2	4 11	6 0	5 7	7 0	29.50

SINGLE SETTING, WITH HALF ARCH FRONT.

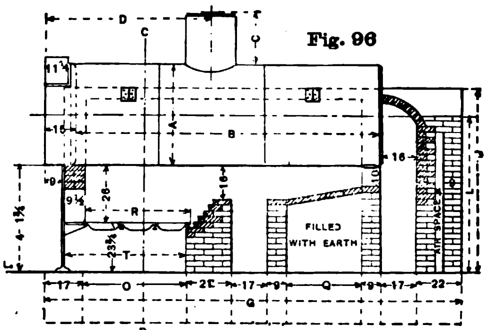
PLAN F.

Fig. 97



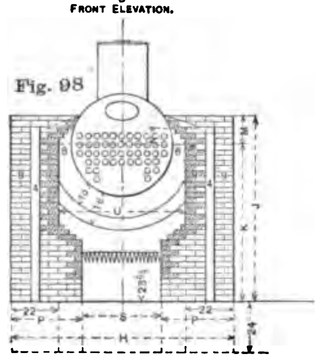
FRONT ELEVATION.

Fig. 96



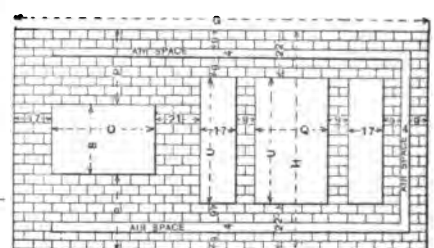
SIDE ELEVATION SECTION THROUGH A. B.

Fig. 98



SECTION THROUGH C. D.

Fig. 99



GROUND PLAN.

SPECIFICATIONS FOR SINGLE SETTING.—PLAN F.

Nom'l Rated Horse Power.	Shell. Diam In.	Shell. Lgth Ft.	Depth Smoke Box, Inches.	Thickness of Boiler Plate. Shell In.	Thickness of Boiler Plate. Heads In.	Boilerplate Homogeneous Steel. Tensile Strength per Square Inch.	Safe Working Pressure for Boiler per Square Inch. Longitudinal Seams, Double Rivet'd	Dome. Diam In.	Dome. Hght In.	Size of Steam Outlet, Inches.	Tubes. No.	Tubes. Diam In.	Tubes. Lgth Ft.	Total Heating Surface, Square Feet.	Stack. Diam In.	Stack. Lgth Ft.	Shipping Weight. Pounds Approximate. Boiler Only.	Fixtures Only.	Boiler and Fixtures.
25	36	12	15	$\frac{1}{4}$	$\frac{3}{8}$	60,000 lbs.	166 lbs.	18	20	$2\frac{1}{2}$	26	3	12	318	18	35	2,970	1,830	4,800
30	40	12	15	$\frac{1}{4}$	$\frac{3}{8}$	60,000 "	150 "	20	22	$2\frac{1}{2}$	34	3	12	404	20	40	3,441	2,159	5,600
35	42	12	15	$\frac{1}{4}$	$\frac{7}{16}$	60,000 "	143 "	20	22	$2\frac{1}{2}$	40	3	12	464	22	40	3,760	2,240	6,000
40	46	12	15	$\frac{1}{4}$	$\frac{7}{16}$	60,000 "	130 "	24	24	3	42	3	12	491	22	40	4,345	2,355	6,700
45	48	12	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 "	140 "	24	24	3	48	3	12	551	24	40	5,045	2,355	7,400
55	52	14	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 "	130 "	28	26	$3\frac{1}{2}$	44	$3\frac{1}{2}$	14	693	26	40	6,620	2,880	9,500
60	54	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	$3\frac{1}{2}$	46	$3\frac{1}{2}$	14	721	26	40	7,300	3,000	10,300
70	54	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	$3\frac{1}{2}$	40	4	16	817	26	40	8,711	3,089	11,800
75	60	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	4	62	$3\frac{1}{2}$	14	940	30	40	9,180	3,420	12,600
85	60	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	$4\frac{1}{2}$	52	4	16	1,045	30	40	10,643	3,457	14,100
100	66	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	136 "	32	32	$4\frac{1}{2}$	64	4	16	1,265	36	40	13,294	4,206	17,500
125	72	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	125 "	32	32	5	82	4	16	1,578	40	40	15,900	4,700	20,600

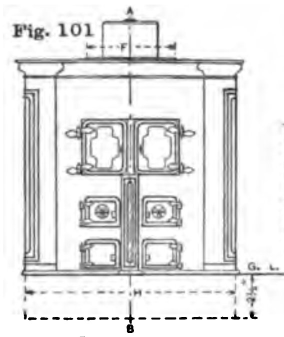
DIMENSIONS FOR SINGLE SETTING.—PLAN F.

Nominal Rated Horse Power of Boiler.	Boilers. A Diameter of Boiler.	B Length of Boiler.	C Top of Boiler to Top of Dome Flange.	D Center of Independent Dome to Front Boiler.	E Centerof Riveted Dome to Front of Boiler.	F Width of Stack Saddle.	Setting. G Length of Foundation.	H Width of Foundation.	I Floor Line to Top of Safety Valve.	J Heigh of Walls.	K Floor Line to Center of Boiler, Front End.	L Floor Line to Center of Boiler, Back End.	M Center of Boiler to Top of Walls.	N Center of Boiler to Closing in of Side Walls.	O Front Wall to Bottom of Bridge Wall.	P Thickness of Jamb.	Q Length of Apron.	Furnace. R Length of Furnace.	S Width of Furnace.	T Length of Ash Pits.	U Width of Soot and Back Pit.	Number of Sq. Feet in Grate Surface.
	IN.	FT.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	SQ. FT.
25	36	12	21	7 2	6 6	1 8	16 4	7 6	10 4	6 7	5 8	5 7	11	6	2 9	2 1	4 3	2 10	3 4	3 6	3 10	9.44
30	40	12	23	7 2	6 6	2 2	16 4	7 10	10 10	6 10	5 10	5 9	12	7	3 3	2 3	3 9	3 4	3 4	4 0	4 2	11.11
35	42	12	23	7 2	6 6	2 8	16 4	8 0	11 0	6 11	5 11	5 10	12	7	3 7	2 3	3 5	3 8	3 6	4 4	4 4	12.83
40	46	12	25	7 2	6 6	2 8	16 4	8 4	11 7	7 1	6 1	6 0	12	7	4 1	2 5	2 11	4 2	3 6	4 10	4 8	14.58
45	48	12	25	7 2	6 6	3 6	16 4	8 6	11 9	7 2	6 2	6 1	12	7	4 1	2 5	2 11	4 2	3 8	4 10	4 10	15.28
55	52	14	27	8 7	7 6	3 9	18 4	9 0	12 3	7 6	6 4	6 3	14	9	4 1	2 6	4 11	4 2	4 0	4 10	5 4	16.66
60	54	14	27	8 7	7 6	3 9	18 4	9 2	12 5	7 6	6 5	6 4	13	8	4 1	2 5	4 11	4 2	4 4	4 10	5 6	18.05
70	54	16	27	9 $10\frac{1}{2}$	8 6	3 9	20 4	9 2	12 5	7 6	6 5	6 4	13	8	4 1	2 1	6 11	4 2	5 0	4 10	5 6	20.83
75	60	14	33	8 7	7 6	4 7	18 4	9 8	13 6	7 9	6 8	6 7	13	8	4 1	2 3	4 11	4 2	5 2	4 10	6 0	21.52
85	60	16	33	9 $10\frac{1}{2}$	8 6	4 7	20 4	9 8	13 6	8 1	6 8	6 7	17	12	4 1	2 2	6 11	4 2	5 4	4 10	6 0	22.22
100	66	16	33	9 $10\frac{1}{2}$	8 6	5 0	20 4	10 2	14 4	8 5	6 11	6 10	18	13	4 7	2 5	6 5	4 8	5 4	5 4	6 6	24.88
125	72	16	33	9 $10\frac{1}{2}$	8 6	5 6	20 4	10 8	14 7	8 6	7 2	7 1	16	11	4 10	2 4	6 2	4 11	6 0	5 7	7 0	29.50

SINGLE SETTING, WITH FULL ARCH FRONT.

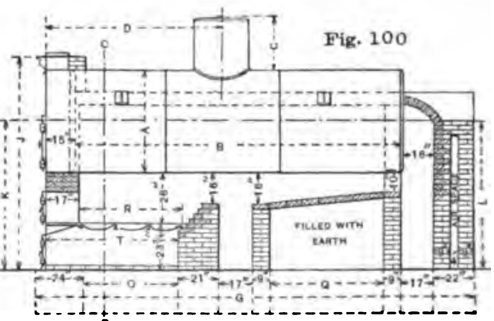
Plan G.

Fig. 101



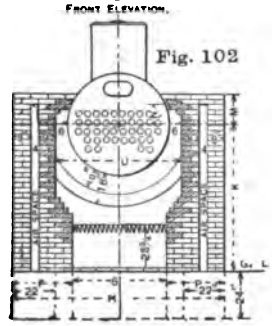
Front Elevation.

Fig. 100



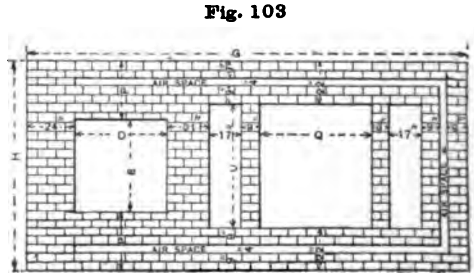
Side Elevation—Section Through A. B.

Fig. 102



Section Through C. D.

Fig. 103



Ground Plan.

SPECIFICATIONS FOR SINGLE SETTING.—PLAN G.

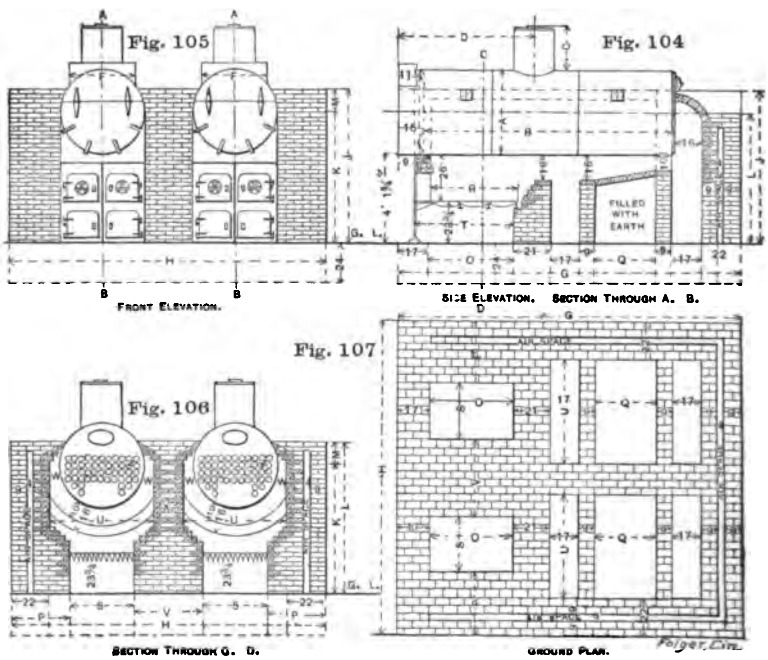
Nom'l Rated Horse Power.	SHELL.		Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE.		BOILER PLATE Homogeneous Steel.	Safe Working Pressure for Boiler per Square Inch. Longitudinal Seams, Double Rivet'd	DOME.		Size of Steam Outlet, Inches.	TUBES.			Total Heating Surface, Square Feet.	STACK.		SHIPPING WEIGHT. Pounds Approximate.		
	Diam In.	Lgth Ft.		Shell In.	Heads In.	Tensile Strength per Square Inch.		Diam In.	Hght In.		No.	Diam In.	Lgth Ft.		Diam In.	Lgth Ft.	Boiler Only.	Fixtures Only.	Boiler and Fixtures.
25	36	12	15	$\frac{1}{4}$	$\frac{3}{8}$	60,000 lbs.	166 lbs.	18	20	$2\frac{1}{2}$	26	3	12	318	18	35	2,970	1,830	4,800
30	40	12	15	$\frac{1}{4}$	$\frac{3}{8}$	60,000 "	150 "	20	22	$2\frac{1}{2}$	34	3	12	404	20	40	3,441	2,159	5,600
35	42	12	15	$\frac{1}{4}$	$\frac{7}{16}$	60,000 "	143 "	20	22	$2\frac{1}{2}$	40	3	12	464	22	40	3,760	2,240	6,000
40	46	12	15	$\frac{1}{4}$	$\frac{7}{16}$	60,000 "	130 "	24	24	3	42	3	12	491	22	40	4,345	2,355	6 700
45	48	12	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 "	140 "	24	24	3	48	3	12	551	24	40	5,045	2,355	7,400
55	52	14	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 "	130 "	28	26	$3\frac{1}{2}$	44	$3\frac{1}{2}$	14	693	26	40	6,620	2,880	9,500
60	54	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	$3\frac{1}{2}$	46	$3\frac{1}{2}$	14	721	26	40	7,300	3,000	10,300
70	54	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	$3\frac{1}{2}$	40	4	16	817	26	40	8,711	3,089	11,800
75	60	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	4	62	$3\frac{1}{2}$	14	940	30	40	9,180	3,420	12,600
85	60	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	$4\frac{1}{2}$	52	4	16	1,045	30	40	10,643	3,457	14,100
100	66	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	136 "	32	32	$4\frac{1}{2}$	64	4	16	1,265	36	40	13,294	4,206	17,500
125	72	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	125 "	32	32	5	82	4	16	1,578	40	40	15,900	4,700	20,600

DIMENSIONS FOR SINGLE SETTING.—PLAN G.

	BOILERS.						SETTING.											FURNACE.				
Nominal Rated Horse Power of Boiler.	Diameter of Boiler.	Length of Boiler.	Top of Boiler to Top of Dome Flange.	Center of Independent Dome to Front Boiler.	Center of Riveted Dome to Front of Boiler.	Width of Stack Saddle.	Length of Foundation.	Width of Foundation.	Floor Line to Top of Safety Valve.	Height of Walls.	Floor Line to Center of Boiler, Front End.	Floor Line to Center of Boiler, Back End.	Center of Boiler to Top of Walls.	Center of Boiler to Closing in of Side Walls.	Front Wall to Bottom of Bridge Wall.	Thickness of Jamb.	Length of Apron.	Length of Furnace.	Width of Furnace.	Length of Ash Pit.	Width of Soot and Back Pits.	Number of Sq. Feet in Grate Surface.
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	
	IN.	FT.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	SQ. FT.
25	36	12	21	7 2	6 6	1 8	17 0	7 6	10 4	8 4	5 8	5 7	11	6	2 9	2 1	4 4	2 10	3 4	4 2	3 10	9.44
30	40	12	23	7 2	6 6	2 2	17 0	7 10	10 10	8 4	5 10	5 9	12	7	3 3	2 3	3 10	3 4	3 4	4 8	4 2	11.11
35	42	12	23	7 2	6 6	2 8	17 0	8 0	11 0	8 4	5 11	5 10	12	7	3 7	2 3	3 6	3 8	3 6	5 0	4 4	12.83
40	46	12	25	7 2	6 6	2 8	17 0	8 4	11 7	9 2	6 1	6 0	12	7	4 1	2 5	3 0	4 2	3 6	5 6	4 8	14.58
45	48	12	25	7 2	6 6	3 6	17 0	8 6	11 9	9 2	6 2	6 1	12	7	4 1	2 5	3 0	4 2	3 8	5 6	4 10	15.28
55	52	14	27	8 7	7 6	3 9	19 0	9 0	12 3	9 2	6 4	6 3	14	9	4 1	2 6	5 0	4 2	4 0	5 6	5 4	16.66
60	54	14	27	8 7	7 6	3 9	19 0	9 2	12 5	10 2	6 5	6 4	13	8	4 1	2 5	5 0	4 2	4 4	5 6	5 6	18.05
70	54	16	27	9 $10\frac{1}{2}$	8 6	3 9	21 0	9 2	12 5	10 2	6 5	6 4	13	8	4 1	2 1	7 0	4 2	5 0	5 6	5 6	20.83
75	60	14	33	8 7	7 6	4 7	19 0	9 8	13 6	10 2	6 8	6 7	13	8	4 1	2 3	5 0	4 2	5 2	5 6	6 0	21.52
85	60	16	33	9 $10\frac{1}{2}$	8 6	4 7	21 0	9 8	13 6	10 2	6 8	6 7	17	12	4 1	2 2	7 0	4 2	5 4	5 6	6 0	22.22
100	66	16	33	9 $10\frac{1}{2}$	8 6	5 0	21 0	10 2	14 4	11 2	6 11	6 10	18	13	4 7	2 5	6 6	4 8	5 4	6 0	6 6	24.88
125	72	16	33	9 $10\frac{1}{2}$	8 6	5 6	21 0	10 8	14 7	11 2	7 2	7 1	16	11	4 10	2 4	6 3	4 11	6 0	6 3	7 0	29.50

DOUBLE SETTING, WITH HALF ARCH FRONT.

PLAN H.



FRONT ELEVATION. SIDE ELEVATION. SECTION THROUGH A. B.

SECTION THROUGH C. D. GROUND PLAN.

SPECIFICATIONS FOR DOUBLE SETTING.—PLAN H.

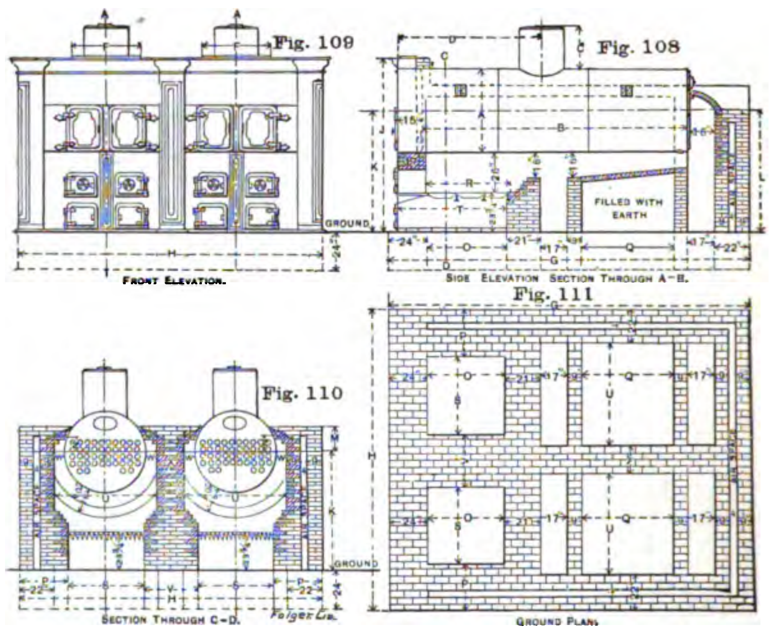
Nom'l Rated Horse Power.	SHELL. Diam In.	SHELL. Lgth Ft.	Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE. Shell In.	THICKNESS OF BOILER PLATE. Heads In.	BOILER PLATE Homogeneous Steel. Tensile Strength per Square Inch.	Safe Working Pressure for Boiler per Square Inch, Longitudinal Seams, Double Rivet'd	DOME. Diam In.	DOME. Hght In.	Size of Steam Outlet, Inches.	TUBES. No.	TUBES. Diam In.	TUBES. Lgth Ft.	Total Heating Surface, Square Feet.	STACK. Diam In.	STACK. Lgth Ft.	SHIPPING WEIGHT. Pounds Approximate. Boilers Only.	SHIPPING WEIGHT. Fixtures Only.	SHIPPING WEIGHT. Boiler and Fixtures.
110	52	14	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 lbs.	130 lbs.	28	26	5	44	$3\frac{1}{2}$	14	1,386	34	40	13,240	6,160	19,400
120	54	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	5	46	$3\frac{1}{2}$	14	1,442	36	40	14,600	6,500	21,100
140	54	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	139 "	28	26	5	40	4	16	1,634	36	40	17,422	6,678	24,100
150	60	14	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	6	62	$3\frac{1}{2}$	14	1,880	40	40	18,360	7,340	25,700
170	60	16	15	$\frac{5}{16}$	$\frac{1}{2}$	60,000 "	126 "	32	32	7	52	4	16	2,090	40	40	21,286	7,414	28,700
200	66	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	136 "	32	32	7	64	4	16	2,530	44	40	26,588	8,412	35,000
250	72	16	15	$\frac{3}{8}$	$\frac{1}{2}$	60,000 "	125 "	32	32	7	82	4	16	3,156	48	40	31,800	10,000	41,800

DIMENSIONS FOR DOUBLE SETTING.—PLAN H.

Nominal Rated Horse Power of Boilers.	BOILERS. Diameter of Boilers. A (In.)	Length of Boilers. B (Ft.)	Top of Boilers to Top of Dome Flanges. C (In.)	Center of Independent Domes to Front of Boilers. D (Ft. In.)	Center of Riveted Domes to Front of Boilers. E (Ft. In.)	Width of Stack Saddles. F (Ft. In.)	SETTING. Length of Foundation. G (Ft. In.)	Width of Foundation. H (Ft. In.)	Floor Line to Top of Safety Valves. I (Ft. In.)	Height of Walls. J (Ft. In.)	Floor Line to Center of Boilers, Front End. K (Ft. In.)	Floor Line to Center of Boilers, Back End. L (Ft. In.)	Center of Boilers to Top of Walls. M (In.)	Center of Boilers to Closing in of Side Walls. N (In.)	Front Walls to Bottom of Bridge Walls. O (Ft. In.)	Thickness of Outside Jambs. P (Ft. In.)	Length of Aprons. Q (Ft. In.)	FURNACE. Length of Furnaces. R (Ft. In.)	Width of Furnaces. S (Ft. In.)	Length of Ash Pits. T (Ft. In.)	Width of Soot and Back Pits. U (Ft. In.)	Thickness of Middle Jamb. V (Ft. In.)	Space Between Boilers and Sides of Furnace. W (In.)	Thickness of Walls Between Boilers. X (In.)	Number of Square Feet in Grate Surface. (Sq. Ft.)
110	52	14	27	8 7	7 6	3 9	18 4	15 9	12 3	7 6	6 4	6 3	14	9	4 1	2 6	4 11	4 2	4 0	4 10	5 4	2 9	6	17	33.32
120	54	14	27	8 7	7 6	3 9	18 4	16 1	12 5	7 6	6 5	6 4	13	8	4 1	2 5	4 11	4 2	4 4	4 10	5 6	2 7	6	17	36.10
140	54	16	27	9 $10\frac{1}{2}$	8 6	3 9	20 4	16 1	12 5	7 6	6 5	6 4	13	8	4 1	2 1	6 11	4 2	5 0	4 10	5 6	1 11	6	17	41.66
150	60	14	33	8 7	7 6	4 7	18 4	17 1	13 6	7 9	6 8	6 7	13	8	4 1	2 3	4 11	4 2	5 2	4 10	6 0	2 3	6	17	43.04
170	60	16	33	9 $10\frac{1}{2}$	8 6	4 7	20 4	17 1	13 6	8 1	6 8	6 7	17	12	4 1	2 2	6 11	4 2	5 4	4 10	6 0	2 1	6	17	44.44
200	66	16	33	9 $10\frac{1}{2}$	8 6	5 0	20 4	18 1	14 4	8 5	6 11	6 10	18	13	4 7	2 5	6 5	4 8	5 4	5 4	6 6	2 7	6	17	49.76
250	72	16	33	9 $10\frac{1}{2}$	8 6	5 6	20 4	19 1	14 7	8 6	7 2	7 1	16	11	4 10	2 4	6 2	4 11	6 0	5 7	7 0	2 5	6	17	59.00

DOUBLE SETTING, WITH FULL FRONT.

PLAN I.



SPECIFICATIONS FOR DOUBLE SETTING.—PLAN I.

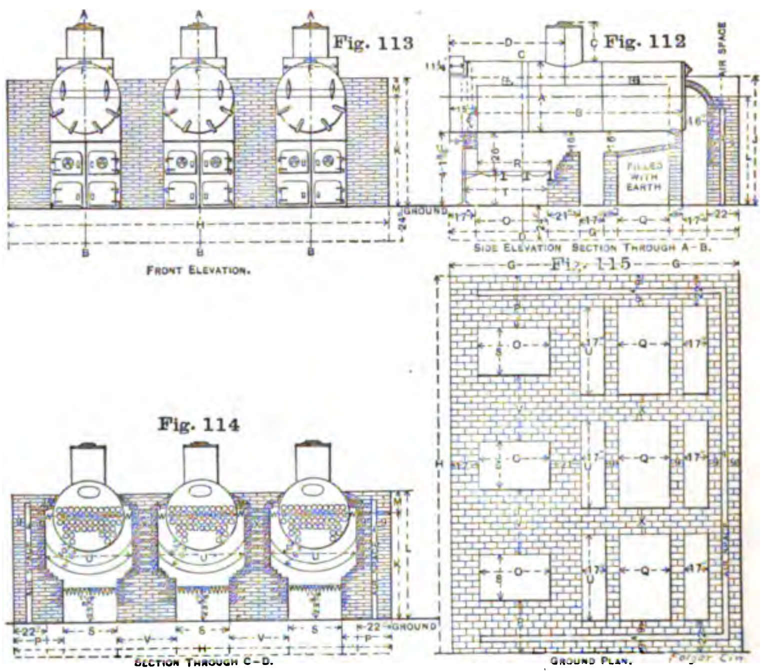
Nom'l Rated Horse Power	SHELL.		Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE.		BOILER PLATE Homogeneous Steel.	Safe Working Pressure for Boiler per Square Inch. Longitudinal Seams, Double Rivet'd	DOME.		Size of Steam Outlet, Inches.	TUBES.			Total Heating Surface, Square Feet.	STACK.		SHIPPING WEIGHT. Pounds Approximate.		
	Diam In.	Lgth Ft.		Shell In.	Heads In.	Tensile Strength per Square Inch.		Diam In.	Hght In.		No.	Diam In.	Lgth Ft.		Diam In.	Lgth Ft.	Boilers Only.	Fixtures Only.	Boiler and Fixtures.
110	52	14	15	9/32	7/16	60,000 lbs.	130 lbs.	28	26	5	44	3½	14	1,386	34	40	13,240	6,160	19,400
120	54	14	15	5/16	½	60,000 "	139 "	28	26	5	46	3½	14	1,442	36	40	14,600	6,500	21,100
140	54	16	15	5/16	½	60,000 "	139 "	28	26	5	40	4	16	1,634	36	40	17,422	6,678	24,100
150	60	14	15	5/16	½	60,000 "	126 "	32	32	6	62	3½	14	1,880	40	40	18,360	7,340	25,700
170	60	16	15	5/16	½	60,000 "	126 "	32	32	7	52	4	16	2,090	40	40	21,286	7,414	28,700
200	66	16	15	⅜	½	60,000 "	136 "	32	32	7	64	4	16	2,530	44	40	26,588	8,412	35,000
250	72	16	15	⅜	½	60,000 "	125 "	32	32	7	82	4	16	3,156	48	40	31,800	10,000	41,800

DIMENSIONS FOR DOUBLE SETTING.—PLAN I.

	BOILERS.						SETTING.											FURNACE							
Nominal Rated Horse Power of Boilers.	Diameter of Boilers.	Length of Boilers.	Top of Boilers to Top of Dome Flanges.	Center of Independent Domes to Front of Boilers.	Center of Riveted Domes to Front of Boilers.	Width of Stack Saddles.	Length of Foundation.	Width of Foundation.	Floor Line to Top of Safety Valves.	Floor Line to Top of Fire Fronts.	Floor Line to Center of Boilers, Front End.	Floor Line to Center of Boilers, Back End.	Center of Boilers to Top of Walls.	Center of Boilers to Closing in of Side Walls.	Front Walls to Bottom of Bridge Walls.	Thickness of Outside Jambs.	Length of Aprons.	Length of Furnaces.	Width of Furnaces.	Length of Ash Pits.	Width of Soot and Back Pits.	Thickness of Middle Jamb.	Space Between Boilers and Sides of Furnace.	Thickness of Walls Between Boilers.	Number of Square Feet in Grate Surface.
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	
	IN.	FT.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	SQ. FT.
110	52	14	27	8 7	7 6	3 9	19 0	15 9	12 3	9 2	6 4	6 3	14	9	4 1	2 6	5 0	4 2	4 0	5 6	5 4	2 9	6	17	33.32
120	54	14	27	8 7	7 6	3 9	19 0	16 1	12 5	10 2	6 5	6 4	13	8	4 1	2 5	5 0	4 2	4 4	5 6	5 6	2 7	6	17	36.10
140	54	16	27	9 10½	8 6	3 9	21 0	16 1	12 5	10 2	6 5	6 4	13	8	4 1	2 1	7 0	4 2	5 0	5 6	5 6	1 11	6	17	41.66
150	60	14	33	8 7	7 6	4 7	19 0	17 1	13 6	10 2	6 8	6 7	13	8	4 1	2 3	5 0	4 2	5 2	5 6	6 0	2 3	6	17	43.04
170	60	16	33	9 10½	8 6	4 7	21 0	17 1	13 6	10 2	6 8	6 7	17	12	4 1	2 2	7 0	4 2	5 4	5 6	6 0	2 1	6	17	44.44
200	66	16	33	9 10½	8 6	5 0	21 0	18 1	14 4	11 2	6 11	6 10	18	13	4 7	2 5	6 6	4 8	5 4	6 0	6 6	2 7	6	17	49.76
250	72	16	33	9 10½	8 6	5 6	21 0	19 1	14 7	11 2	7 2	7 1	16	11	4 10	2 4	6 3	4 11	6 0	6 3	7 0	2 5	6	17	59.00

TRIPLE SETTING, WITH HALF ARCH FRONT.

PLAN J.



SPECIFICATIONS FOR TRIPLE SETTING.—PLAN J.

Nom'l Rated Horse Power.	SHELL.		Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE.		BOILERPLATE Homogeneous Steel.	Safe Working Pressure for Boiler per Square Inch. Longitudinal Seams, Double Rivet'd	DOME.		Size of Steam Outlet, Inches.	TUBES.			Total Heating Surface, Square Feet.	STACK.		SHIPPING WEIGHT. Pounds Approximate.		
	Diam In.	Lgth Ft.		Shell In.	Heads In.	Tensile Strength per Square Inch.		Diam In.	Hght In.		No.	Diam In.	Lgth Ft.		Diam In.	Lgth Ft.	Boilers Only.	Fixtures Only.	Boilers and Fixtures.
165	52	14	15	9/32	7/16	60,000 lbs.	130 lbs.	28	26	6	44	3½	14	2,079	40	40	19,860	8,740	28,600
180	54	14	15	5/16	½	60,000 "	139 "	28	26	6	46	3½	14	2,163	42	40	21,900	9,200	31,100
210	54	16	15	5/16	½	60,000 "	139 "	28	26	6	40	4	16	2,451	42	40	26,133	9,467	35,600
225	60	14	15	5/16	½	60,000 "	126 "	32	32	7	62	3½	14	2,820	46	40	27,540	10,760	38,300
255	60	16	15	5/16	½	60,000 "	126 "	32	32	8	52	4	16	3,135	46	40	31,929	10,871	42,800
300	66	16	15	3/8	½	60,000 "	136 "	32	32	8	64	4	16	3,795	50	40	39,882	12,518	52,400
375	72	16	15	3/8	½	60,000 "	125 "	32	32	9	82	4	16	4,734	54	40	47,700	14,000	61,700

DIMENSIONS FOR TRIPLE SETTING.—PLAN J.

Nominal Rated Horse Power of Boilers.	BOILERS. Diameter of Boilers.	Length of Boilers.	Top of Boilers to Top of Dome Flanges.	Center of Independent Domes to Front of Boilers.	Center of Riveted Domes to Front of Boilers.	Width of Stack Saddles.	SETTING. Length of Foundation.	Width of Foundation.	Floor Line to Top of Safety Valves.	Height of Walls.	Floor Line to Center of Boilers, Front End.	Floor Line to Center of Boilers, Back End.
	A	B	C	D	E	F	G	H	I	J	K	L
	IN.	FT.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.
165	52	14	27	8 7	7 6	3 9	18 4	22 6	12 3	7 6	6 4	6 3
180	54	14	27	8 7	7 6	3 9	18 4	23 0	12 5	7 6	6 5	6 4
210	54	16	27	9 10½	8 6	3 9	20 4	23 0	12 5	7 6	6 5	6 4
225	60	14	33	8 7	7 6	4 7	18 4	24 6	13 6	7 9	6 8	6 7
255	60	16	33	9 10½	8 6	4 7	20 4	24 6	13 6	8 1	6 8	6 7
300	66	16	33	9 10½	8 6	5 0	20 4	26 0	14 4	8 5	6 11	6 10
375	72	16	33	9 10½	8 6	5 6	20 4	27 6	14 7	8 6	7 2	7 1

Nominal Rated Horse Power of Boilers.	SETTING. Center of Boilers to Top of Walls.	Center of Boilers to Closing in of Side Walls.	Front Walls to Bottom of Bridge Walls.	Thickness of Outside Jambs.	Length of Aprons.	FURNACE. Length of Furnaces.	Width of Furnaces.	Length of Ash Pits.	Width of Soot and Back Pits.	Thickness of Middle Jambs.	Space Between Boilers and Sides of Furnace.	Thickness of Walls Between Boilers.	Number of Square Feet in Grate Surface.
	M	N	O	P	Q	R	S	T	U	V	W	X	
	IN.	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	SQ. FT.
165	14	9	4 1	2 6	4 11	4 2	4 0	4 10	5 4	2 9	6	17	49.98
180	13	8	4 1	2 5	4 11	4 2	4 4	4 10	5 6	2 7	6	17	54.15
210	13	8	4 1	2 1	6 11	4 2	5 0	4 10	5 6	1 11	6	17	62.49
225	13	8	4 1	2 3	4 11	4 2	5 2	4 10	6 0	2 3	6	17	64.56
255	17	12	4 1	2 2	6 11	4 2	5 4	4 10	6 0	2 1	6	17	66.66
300	18	13	4 7	2 5	6 5	4 8	5 4	5 4	6 6	2 7	6	17	74.64
375	16	11	4 10	2 4	6 2	4 11	6 0	5 7	7 0	2 5	6	17	88.50

TRIPLE SETTING, WITH FULL FRONT.

PLAN K.

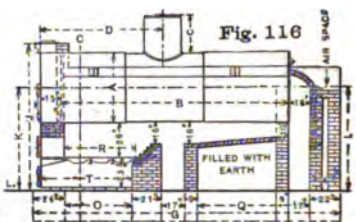
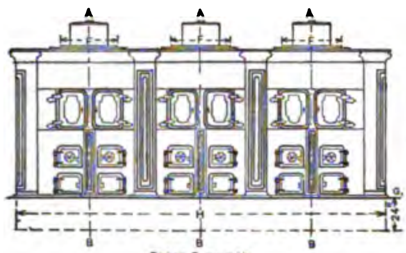


Fig. 116

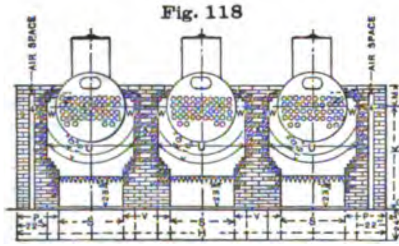
SIDE ELEVATION SECTION THROUGH A-B.



FRONT ELEVATION.

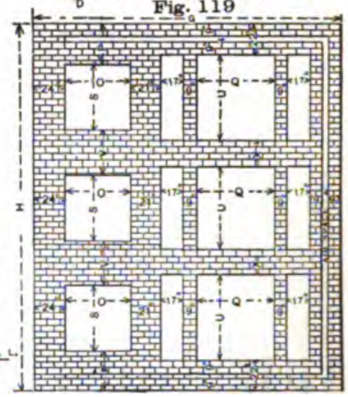
Fig. 117

Fig. 118



SECTION THROUGH C-D.

Fig. 119



GROUND PLAN.

SPECIFICATIONS FOR TRIPLE SETTING.—PLAN K.

Nom'l Rated Horse Power.	SHELL.		Depth Smoke Box, Inches.	THICKNESS OF BOILER PLATE.		BOILERPLATE Homogeneous Steel.	Safe Working Pressure for Boiler per Square Inch.	DOME.		Size of Steam Outlet, Inches.	TUBES.			Total Heating Surface, Square Feet.	STACK.		SHIPPING WEIGHT. Pounds Approximate.		
	Diam In.	Lgth Ft.		Shell In.	Heads In.	Tensile Strength per Square Inch.	Longitudinal Seams, Double Rivet'd	Diam In.	Hght In.		No.	Diam In.	Lgth Ft.		Diam In.	Lgth Ft.	Boilers Only.	Fixtures Only.	Boiler and Fixtures.
165	52	14	15	$\frac{9}{32}$	$\frac{7}{16}$	60,000 lbs.	130 lbs.	28	26	6	44	3½	14	2,079	40	40	19,860	8,740	28,600
180	54	14	15	$\frac{5}{16}$	½	60,000 "	139 "	28	26	6	46	3½	14	2,163	42	40	21,900	9,200	31,100
210	54	16	15	$\frac{5}{16}$	½	60,000 "	139 "	28	26	6	40	4	16	2,451	42	40	26,133	9,467	35,600
225	60	14	15	$\frac{5}{16}$	½	60,000 "	126 "	32	32	7	62	3½	14	2,820	46	40	27,540	10,760	38,300
255	60	16	15	$\frac{5}{16}$	½	60,000 "	126 "	32	32	8	52	4	16	3,135	46	40	31,929	10,871	42,800
300	66	16	15	⅜	½	60,000 "	136 "	32	32	8	64	4	16	3,795	50	40	39,882	12,518	52,400
375	72	16	15	⅜	½	60,000 "	125 "	32	32	9	82	4	16	4,734	54	40	47,700	14,000	61,700

DIMENSIONS FOR TRIPLE SETTING.—PLAN K.

Nominal Rated Horse Power of Boilers.	BOILERS. Diameter of Boilers.	Length of Boilers.	Top of Boilers to Top of Dome Flanges.	Center of Independent Domes to Front of Boilers	Center of Riveted Domes to Front of Boilers.	Width of Stack Saddles.	SETTING. Length of Foundation.	Width of Foundation.	Floor Line to Top of Safety Valves.	Floor Line to Top of Fire Fronts.	Floor Line to Center of Boilers, Front End.	Floor Line to Center of Boilers, Back End.
	A	B	C	D	E	F	G	H	I	J	K	L
	IN.	FT	IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.
165	52	14	27	8 7	7 6	3 9	19 0	22 6	12 3	9 2	6 4	6 3
180	54	14	27	8 7	7 6	3 9	19 0	23 0	12 5	10 2	6 5	6 4
210	54	16	27	9 10½	8 6	3 9	21 0	23 0	12 5	10 2	6 5	6 4
225	60	14	33	8 7	7 6	4 7	19 0	24 6	13 6	10 2	6 8	6 7
255	60	16	33	9 10½	8 6	4 7	21 0	24 6	13 6	10 2	6 8	6 7
300	66	16	33	9 10½	8 6	5 0	21 0	26 0	14 4	11 2	6 11	6 10
375	72	16	33	9 10½	8 6	5 6	21 0	27 6	14 7	11 2	7 2	7 1

Nominal Rated Horse Power of Boilers.	SETTING. Center of Boilers to Top of Walls.	Center of Boilers to Closing in of Side Walls.	Front Walls to Bottom of Bridge Walls.	Thickness of Outside Jambs.	Length of Aprons.	FURNACE. Length of Furnaces.	Width of Furnaces.	Length of Ash Pits.	Width of Soot and Back Pits.	Thickness of Middle Jambs.	Space Between Boilers and Sides of Furnace.	Thickness of Walls Between Boilers.	Number of Square Feet in Grate Surface.
	M	N	O	P	Q	R	S	T	U	V	W	X	
	IN.	IN.	FT. IN	FT. IN	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	FT. IN.	IN.	IN.	SQ. FT.
165	14	9	4 1	2 6	5 0	4 2	4 0	5 6	5 4	2 9	6	17	49.98
180	13	8	4 1	2 5	5 0	4 2	4 4	5 6	5 6	2 7	6	17	54.15
210	13	8	4 1	2 1	7 0	4 2	5 0	5 6	5 6	1 11	6	17	62.49
225	13	8	4 1	2 3	5 0	4 2	5 2	5 6	6 0	2 3	6	17	64.56
255	17	12	4 1	2 2	7 0	4 2	5 4	5 6	6 0	2 1	6	17	66.66
300	18	13	4 7	2 5	6 6	4 8	5 4	6 0	6 6	2 7	6	17	74.64
375	16	11	4 10	2 4	6 3	4 11	6 0	6 3	7 0	2 5	6	17	88.50

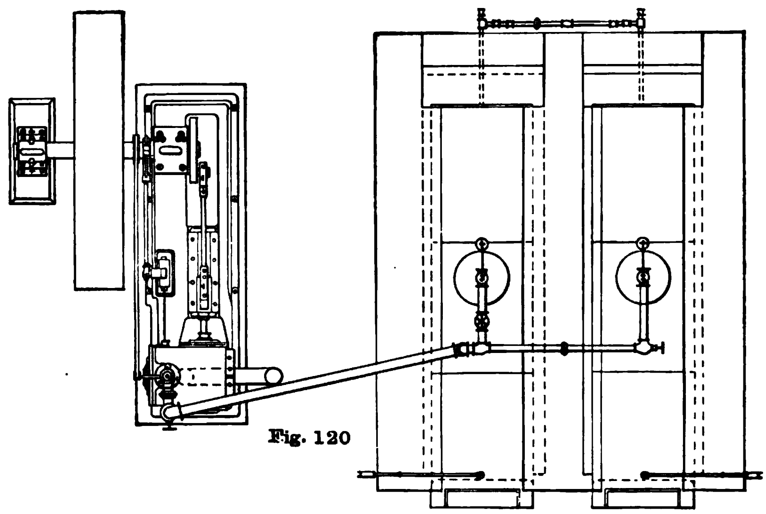


Fig. 120

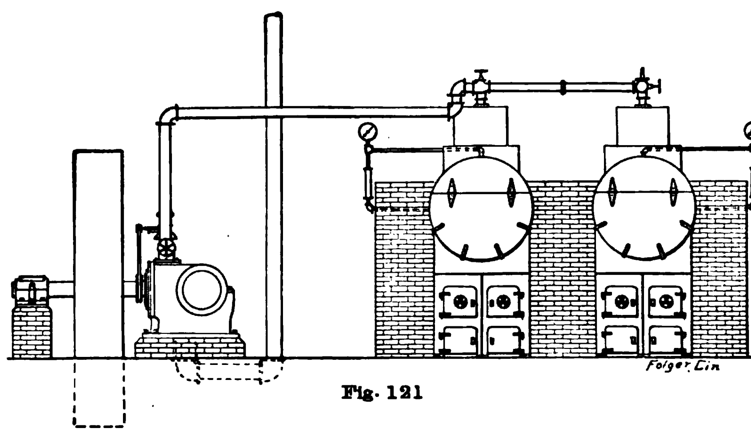


Fig. 121

PLAN AND ELEVATION OF LEFT HAND ENGINE WITH TUBULAR BOILERS.

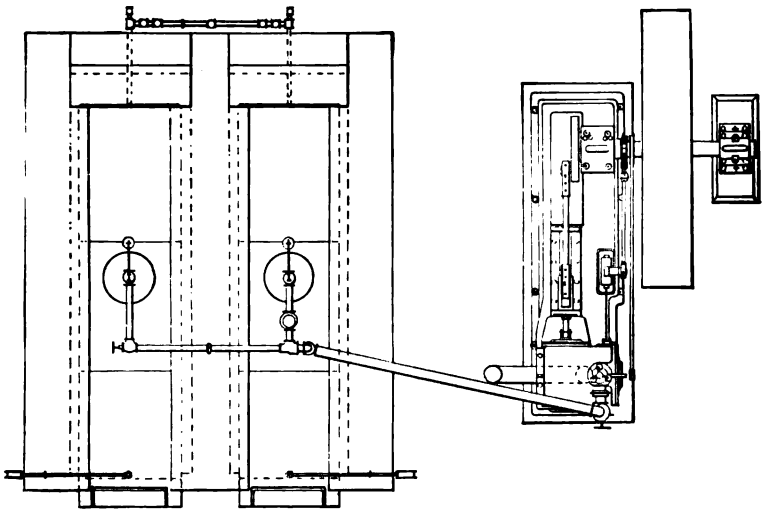


Fig. 122

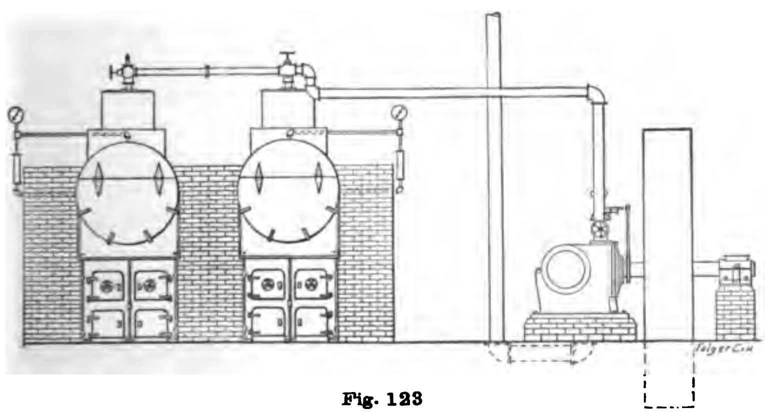


Fig. 123

PLAN AND ELEVATION OF RIGHT HAND ENGINE WITH TUBULAR BOILERS

MATERIALS FOR TUBULAR BOILER SETTINGS.

BOILERS.	SINGLE SETTING.						DOUBLE SETTING.						TRIPLE SETTING.					
	Common Brick.	Fire Brick.	Sand, Bushels.	Cement, Barrels.	Lime, Barrels.	Fire Clay, Pounds.	Common Brick.	Fire Brick.	Sand, Bushels.	Cement, Barrels.	Lime, Barrels.	Fire Clay, Pounds.	Common Brick.	Fire Brick.	Sand, Bushels.	Cement, Barrels.	Lime, Barrels.	Fire Clay, Pounds.
IN. FT.																		
36 x 8	7,600	660	60	7½	3	396	11,850	1,300	95	12	4¾	780	16,000	1,900	130	16	6	1,200
36 x 10	8,600	750	70	8½	3½	450	13,250	1,500	106	13¼	5½	900	18,000	2,200	140	18	7½	1,300
36 x 12	9,800	820	78	10	4	492	15,000	1,650	120	15	6	990	20,000	2,500	160	20	8	1,500
40 x 12	10,000	850	80	10	4	510	15,500	1,690	124	15½	6¼	1,014	20,500	2,500	170	21	8½	1,500
42 x 12	10,300	850	83	10½	4¼	510	16,000	1,700	128	16	6½	1,020	22,000	2,500	175	22	9	1,550
46 x 12	11,000	960	88	11	4½	576	17,300	1,920	138	17½	7	1,152	23,000	2,800	185	24	9½	1,700
48 x 12	11,200	1,000	90	11¼	4½	600	17,700	2,000	142	17¾	7	1.200	24,000	3,000	190	25	9½	1,800
52 x 14	12,300	1,160	98	12½	5	696	19,250	2,320	154	19¼	7¾	1,392	26,000	3,500	210	26	10½	2,000
54 x 14	12,400	1,260	100	12½	5	756	19,500	2,520	156	19½	8	1,512	27,000	3,800	212	27	11	2,300
54 x 16	13,200	1,350	106	13¼	5¼	810	20,500	2,700	164	20½	8¼	1,620	28,000	4,000	220	28	11½	2,450
60 x 14	12,700	1,260	102	12¾	5¼	756	20,000	2,500	160	20	8	1,500	27,500	3,850	218	27½	11	2,300
60 x 16	14,100	1,380	113	14	5½	828	22,000	2,760	176	22	9	1,656	30,000	4,100	230	30	12½	2,500
66 x 16	14,850	1,680	119	15	6	1,008	23,400	3,360	187	23½	9½	2,016	32,000	5,000	250	32	13	3,000
72 x 16	15,250	1,750	122	15¼	6¼	1,050	24,250	3,500	194	24¼	10	2,100	33,500	5,250	266	34	14	3,200

The estimates of materials in the table, for double and triple setting, provide for placing boilers independently with dividing wall between them.

HOW TO DRAUGHT PLANS AND SPECIFICATIONS FOR TUBULAR BOILERS.

Care must be taken to build the side walls so that the longitudinal seams will be protected against the fire. Therefore, those seams should always be high enough to allow them to be covered without destroying any of the effective heating surface of the boiler.

A careful study of the plans for various styles of settings of boilers, together with the tables and specifications, will enable the engineer of ordinary skill to draught his own plans and specifications. A variety of plans are given for the purpose of enabling engineers who desire to draught their own plans and specifications to select whichever plan may be found most suitable for their purpose.

After having determined upon the plan desired, place a piece of tracing paper over the plan selected, and trace the different figures upon the tracing paper, omitting the letters. When completed, select from the proper tables, the figures that should be placed in the spaces occupied by the letters in the plan and transfer those figures to their proper places in the tracing, and when the places in the tracing, now occupied by letters in the plan, have been filled with the proper figures, the plan will be complete.

By comparing the letters in each of the plans with those at the head of each of the columns in the table belonging to the plan selected, the figures in the table will be readily understood. Before transferring figures to the tracing paper containing the plan selected, the length and diameter of the boiler must first be determined. Then run down the columns A and B until the size of boiler selected is reached, and then transfer the length and diameter to the tracings, taking the plan from which the tracing was made as a guide as to where to place the figures. Having placed the length and diameter in the proper places on the tracing, proceed to transfer the figures in the succeeding columns to the right of the columns containing the length and diameter selected, and continue on the same line to the right until all of the figures representing dimensions have been transferred to the tracing.

The next thing to be done is to determine the tensile strength of the plate per square inch of section of which the boiler is to be constructed. As good material for boilers can be made of homogeneous steel, ranging in tensile strength from 55,000 to 70,000 pounds per square inch, and of iron ranging from 50,000 to 60,000 pounds per square inch, no difficulty need be experienced in making the proper selection, if care is taken to prescribe the proper ductility for the metal selected. Homogeneous steel plates having a tensile strength of from 55,000 to 65,000 pounds per square inch, and iron plates having a tensile strength of from 50,000 to 60,000 pounds per square inch, are considered better than boiler material having greater or less tensile strength.

But whatever tensile strength between the highest and lowest limit named is selected, steel should have a ductility of not less than 50 per cent., and iron plates should have not less than 25 per cent. This means that the test piece, when broken in a testing machine, should show a contraction of area at point of fracture of not less than 50 per cent. for steel, and not less than 25 per cent. for iron.

Having determined upon the diameter and length of boiler; upon the thickness and tensile strength of material; and upon the number of boilers, the engineer is prepared to proceed with the draughting of his specifications. As a guide to construction of boilers, the kind of equipment to provide, attention is called to the specifications at the beginning of this chapter. These, however, can be varied according to circumstances, as they may arise. But the vital points enumerated should be strictly adhered to—that is tensile strength, ductility and thickness of material, and drilling all rivet holes instead of punching, and double riveting all longitudinal seams in every case where high pressure is desired. In other words, whenever a pressure is required to be carried in a boiler, which will produce a strain on the plates exceeding one-sixth of the tensile strength of the plates in the shell, it should be expressly stipulated in the specifications that all rivet holes for the boilers shall be fairly drilled, and that the longitudinal seams shall be double riveted.

STRAIN ON LONGITUDINAL SEAMS.

TO DETERMINE THE STRAIN ON EACH LONGITUDINAL INCH PRODUCED BY ANY GIVEN PRESSURE IN A BOILER.

RULE.—Multiply the given pressure per square inch by one-half of the diameter of the boiler in inches, and the product will give the total strain on each inch in the length of the boiler.

Example.—Let 150 pounds equal given steam pressure per square inch.

Let 40 inches equal given diameter of the boiler.

Then we have:

$$\begin{array}{r} 150 \\ 40 \div 2 = 20 \\ \hline 3000 \text{ lbs.} \end{array}$$

Tending to separate the material.

If the boiler was made of hoops 40 inches in diameter, and 1 inch in width, there would be a strain of 3000 pounds on all parts of each hoop tending to tear it asunder.

Now, suppose that the metal in the hoop was 25 one hundredths of an inch in thickness, and had a tensile strength of 60,000 pounds per square inch, what would be the actual strength of a strip of that material 1 inch in width and 25 one hundredths of an inch in thickness?

To determine the actual strength, the thickness of the material in decimals of an inch is multiplied by the tensile strength in pounds per square inch, and the product will give the actual strength of the material. Thus:

.25
60000

15000.00 lbs. Actual strength of a strip 1 inch in width and 25 one hundredths of an inch in thickness.

Therefore, if it is required to ascertain whether or not the longitudinal seams ought to be double riveted and all rivet holes drilled, we first divide the above answer by 6, and the quotient will give the total strain that ought to be allowed on each longitudinal inch in the boiler for single-riveted longitudinal seams. Thus:

6)15000

2500 lbs. Maximum strain for single riveting.

Now, if we multiply the required steam pressure per square inch by one-half the diameter of the boiler, and find that the product is greater than the quotient in the above example, it will show that the longitudinal seams should be double riveted and all rivet holes in the boiler drilled.

Performing the operation, we have:

150 Pressure per square inch.
20 One-half diameter of boiler.

3000 lbs. Actual strain per longitudinal inch in the boiler.

This demonstrates that single riveting in longitudinal seams will not do. At the same time it may show that even double riveting will not do. If the actual strength of a strip of boiler plate be divided by 5, the quotient will give the total strain allowed on each longitudinal inch for double riveting. Thus:

5) 15000

3000 lbs. One-fifth of actual strength of plate.

In this case, however, it is shown that double-riveted longitudinal seams will enable the boiler (40 inches in diameter, 60,000 pounds tensile strength of material, and 25 one hundredths of an inch thickness of material) to carry a safe-working pressure of 150 pounds per square inch. But if the quotient, as above shown, should be less than the product of the pressure multiplied by one-half of the diameter of the boiler, one of three things will have to be done if the safe-working pressure is to be a fixed amount, as in the above case; either the tensile strength of the material be required to be greater, or the thickness of the material be required to be greater, or else a smaller diameter of boiler will have to be selected.

DIAMETER OF BOILER.

TO DETERMINE THE REQUIRED DIAMETER OF BOILER.

RULE.—Multiply the given thickness of plate in decimals of an inch by the constant 192, *and the product will give required diameter of boiler in inches.*

Example.—Let 375 one thousandths of an inch equal thickness of material.

Let 192 equal a constant.

Then we have:

```
   .375
    192     Constant for double-riveted plate, 60,000
   ----       tensile strength.
    750
  33 75
  37 5
  ------
  72.000 inches.  Required diameter of boiler.
```

HORSE POWER OF BOILER.

In practice, the number of horse powers required in a tubular boiler determines its diameter. For each horse power required the boiler should contain not less than 12 square feet of effective heating surface. Two-thirds of the shell may be exposed to the heated gases of the furnace and be counted as effective heating surface, and two-thirds of the tube surface may also be computed as effective. Therefore, as capacity to do a given or a required amount of work is the first requisite in a boiler, it follows that pressure, diameter and length, with the necessary number of tubes, must be determined before tensile strength and thickness of boiler plate can be determined.

THICKNESS OF PLATE IN SHELL FOR SINGLE RIVETING.

TO DETERMINE THE THICKNESS OF PLATE FOR SHELL OF BOILER WITH SINGLE-RIVETED LONGITUDINAL SEAMS.

RULE.—First, multiply the required steam pressure per square inch by the required diameter of the boiler in inches, then multiply the product by the constant 3, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the required tensile strength of plate per square inch, and the quotient will give the required thickness of plate for single-riveted longitudinal seams.*

Example.—Let 125 pounds equal required steam pressure per square inch.

Let 72 inches equal required diameter of boiler.

Let 3 equal a constant for single riveting.

Let 60,000 pounds equal required tensile strength of plate per square inch.

Then we have: $\frac{125 \times 72 \times 3}{60000} = .45$ Decimals of an inch. Thickness of plate required.

Performing the operation, we have:

125	Steam pressure in pounds.
72	Diameter of boiler in inches.
250	
875	
9000	
3	A constant for single-riveted longitudinal seams.
27000	"Product No. 1."

Next, dividing "Product No. 1" by the tensile strength of plate per square inch, we have:

```
60000 ) 27000.0 ( .45   Decimals of an inch. Thickness of plate required.
        24000 0
        -------
         3000 00
         3000 00
         -------
```

THICKNESS OF PLATE IN SHELL FOR DOUBLE RIVETING.

TO DETERMINE THE THICKNESS OF PLATE FOR SHELL OF BOILER WITH DOUBLE-RIVETED LONGITUDINAL SEAMS.

RULE.—First, multiply the required steam pressure per square inch by the required diameter of the boiler in inches, then multiply the product by the constant 2.5, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the required tensile strength of plate per square inch, and the quotient will give the required thickness of plate for double-riveted longitudinal seams.*

Example.—Let 125 pounds equal required steam pressure per square inch.

Let 72 inches equal required diameter of boiler.

Let 2.5 equal a constant for double riveting.

Let 60,000 pounds equal required tensile strength of plate per square inch.

Then we have: $\frac{125 \times 72 \times 2.5}{60000} = .375$ Decimals of an inch. Thickness of plate required.

Performing the operation, we have:

125	Steam pressure in pounds.
72	Diameter of boiler in inches.
250	
875	
9000	
2.5	A constant for double-riveted longitudinal seams.
45000	
18000	
22500.0	"Product No. 1."

Next, dividing "Product No. 1" by the tensile strength of plate per square inch, we have:

```
60000) 22500.0 (.375   Decimals of an inch. Thickness
       18000 0               of plate required.
       -------
        4500 00
        4200 00
        -------
         300 000
         300 000
         -------
```

TENSILE STRENGTH OF BOILER PLATE FOR SINGLE RIVETING.

TO DETERMINE THE TENSILE STRENGTH OF PLATE FOR SHELL OF BOILER WITH SINGLE-RIVETED LONGITUDINAL SEAMS.

RULE.—First, multiply the required steam pressure per square inch by the required diameter of the boiler in inches, then multiply the product by the constant 3, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the required thickness of plate in decimals of an inch and the quotient will give the required tensile strength, per sectional square inch of plate for single-riveted longitudinal seams.*

Example.—Let 125 pounds equal required steam pressure per square inch.
Let 72 inches equal required diameter of boiler.
Let 3 equal a constant for double riveting.
Let 375 one thousandths of an inch equal required tensile strength per square inch of plate.

Then we have: $\frac{125 \times 72 \times 3}{.375} = 72000$ lbs. Required tensile strength per square inch of plate.

Performing the operation, we have:

```
  125   Steam pressure in pounds.
   72   Diameter of boiler in inches.
  ---
  250
 875
 ----
 9000
    3   A constant for single-riveted longitudinal
 ----     seams.
27000   "Product No. 1."
```

Next, dividing "Product No. 1" by the thickness of plate in decimals of an inch, we have:

```
.375) 27000.000 (72000 lbs.   Required tensile strength
      2625                      per square inch of plate.
      ------
       750
       750
       ---
       000
```

TENSILE STRENGTH OF BOILER PLATE FOR DOUBLE RIVETING.

TO DETERMINE THE TENSILE STRENGTH OF PLATE FOR SHELL OF BOILER WITH DOUBLE-RIVETED LONGITUDINAL SEAMS.

RULE.—First, multiply the required steam pressure per square inch by the required diameter of the boiler in inches, then multiply the product by the constant 2.5, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the required thickness of plate in decimals of an inch, and the quotient will give the required tensile strength of plate per square inch for double-riveted longitudinal seams.*

Example.—Let 125 pounds equal given pressure per square inch.
Let 72 inches equal given diameter of boiler.
Let 2.5 equal a constant.
Let 375 one thousandths of an inch equal required thickness of plate.

Then we have:

$$\frac{125 \times 72 \times 2.5}{.375} = 60000 \text{ lbs. Required tensile strength per square inch of plate.}$$

Performing the operation, we have:

$$\begin{array}{r} 125 \\ 72 \\ \hline 250 \\ 875 \\ \hline 9000 \\ 2.5 \\ \hline 45000 \\ 18000 \\ \hline 22500.0 \end{array} \text{ "Product No. 1."}$$

Then, dividing "Product No. 1" by the thickness of plate, we have:

$$.375\ )\ 22500.000\ (\ 60000 \text{ lbs. Required tensile strength per square inch of plate.}$$
$$\begin{array}{l} 2250 \\ \hline 0\,000 \end{array}$$

Then we have the following data for a boiler:

SINGLE-RIVETED LONGITUDINAL SEAMS.

125 pounds pressure per square inch.
72 inches diameter of boiler.
45 one hundredths of an inch thickness of plate.
60,000 pounds tensile strength per square inch of plate.

Or, we have:

125 pounds pressure per square inch.
72 inches diameter of boiler.
375 one thousandths of an inch thickness of plate.
72,000 pounds tensile strength of material.

In the first set of items it will be observed that the material is too thick, and in the second, that the tensile strength of the material is too great, therefore the holes in the boiler should be drilled and the longitudinal seams double riveted. Hence we select the following data for specifications:

DOUBLE-RIVETED LONGITUDINAL SEAMS.

125 pounds pressure per square inch.
72 inches diameter of boiler.
375 one thousandths of an inch thickness of plate.
60,000 pounds tensile strength per square inch of plate.

The thickness of plate above selected is the least thickness that should be used. Hence, the rule requiring the diameter of the boiler in inches to be divided by the constant 192, gives the minimum thickness of plate that should be used for high-pressure boilers; and dividing the diameter of the boiler in inches by the constant 160, will give the maximum thickness that should be used in boilers to which the heat is applied to the outside of the shell. In no case, however, should the thickness of material in the shell of such boilers exceed ½ inch. Even then, with very hot furnace fires, laps in seams exposed to the fire are liable to burn away and endanger the safety of the boiler.

MAXIMUM THICKNESS OF BOILER PLATE.

TO DETERMINE THE MAXIMUM THICKNESS OF BOILER PLATE FOR BOILERS TO WHICH THE HEAT IS APPLIED TO THE OUTSIDE OF THE SHELL.

RULE.—*Divide the diameter of the boiler in inches by the constant* 160, *and the quotient will give the maximum thickness of material in decimals of an inch.*

Example.—Let 40 inches equal diameter of the boiler.
Let 160 equal a constant.

Then we have:

```
160) 40.0 (0.25 inch.  Maximum thickness of material required.
     32 0
     ----
      8 00
      8 00
      ----
```

CHAPTER XI.

EVAPORATIVE TESTS OF STEAM BOILERS.

HEAT UNITS.

In entering upon the study of this subject, the student should familiarize himself with the meaning of the term "heat unit," as employed in connection with evaporative and calorimeter tests. Many text-book writers seem to regard temperature and heat units as synonymous terms, and use them interchangeably. That is a grave mistake, as there is no similarity between them, because the heat units represent the amount of heat imparted to water or steam in raising either to a certain temperature as shown by a thermometer. For example, it requires 212.9 heat units to raise one pound of water from zero to 212° Fahrenheit, and 965.7 heat units additional to evaporate it to steam at a temperature of 212° Fahrenheit. So while the thermometer registers the sensible heat it does not register the work done or the number of heat units expended in doing the work. In evaporating one pound of water to steam from and at a temperature of 212° Fahrenheit, the thermometer remains stationary in registering 212°; yet in performing the work of evaporating the one pound of water at 212°, 965.7 units of heat are expended; add to that the 212.9 heat units required to raise the water to 212° Fahrenheit from zero, and we have a total of 1178.6 heat units expended in raising water from zero to 212° Fahrenheit, and evaporating it to steam at that temperature. Then, the heat units required to evaporate water to steam from and at any temperature of the water is called latent heat, or heat of vaporization.

The heat unit has been adopted as a standard for measuring work performed by the application of heat, and it represents the amount of heat required to raise one pound of water one degree of the Fahrenheit scale from or at a temperature of 32° Fahrenheit.

It will be seen, by reference to the table of saturated steam, that when the temperature of 212° Fahrenheit has been reached, that the total heat units in one pound of steam increase at the rate of 305 one thousandths of a unit for every degree of temperature above 212° Fahrenheit.

HEAT UNITS IN STEAM.

TO DETERMINE THE TOTAL HEAT UNITS IN ONE POUND OF STEAM EVAPORATED FROM WATER HAVING A TEMPERATURE OVER 212° FAHRENHEIT.

RULE.—Subtract the temperature, 212° *from the indicated temperature and multiply the remainder by* 305 *one thousandths, and then add the total number of heat units in one pound of steam at* 212° (1178.6) *to the product, and the sum will give the total heat of steam for any temperature above* 212°.

Example.—Let 331.805° equal indicated temperature in Fahrenheit degrees.
Let 212° equal temperature of water at boiling point in Fahrenheit degrees.
Let 305 one thousandths of a heat unit equal increase for each degree above 212° Fahrenheit.
Let 1178.6 equal total heat units in one pound of steam at 212° Fahrenheit.

Then we have:

$$[(331.805-212)\times .305]+1178.6=1215.14+$$ Total number of heat units in steam from water at 331.805° Fahrenheit.

Performing the operation, we have:

331.805	Indicated temperature or sensible heat.
212	
119.805	
.305	Increase in heat units for each degree of temperature above 212°
599025	
35 9415	
36.540525	
1178.6	
1215.140525	Total heat units in steam at a temperature of 331.805° Fahrenheit.

FACTOR OF EVAPORATION.

TO DETERMINE THE FACTOR OF EVAPORATION.

RULE.—Subtract the number of heat units in one pound of the feed water due the temperature of that water, from the total number of heat units in one pound of steam due the temperature or pressure of steam under which the test is made, then divide the remainder by 965.7, *and the quotient will give the factor of evaporation.*

Example.—Let 1225.6417 equal total heat units according to the table in one pound of steam at 366.232° Fahrenheit, or at 151.304 pounds pressure by gauge.

Let 62.011 equal heat units in feed water at 62° Fahrenheit.

Let 965.7 equal heat required to evaporate one pound of water from and at 212° Fahrenheit.

Then we have:

$$\frac{1225.6417-62.011}{965.7}=1.2+ \text{ Factor of evaporation.}$$

Performing the operation, we have:

$$\begin{array}{r l}
 & 1225.6417 \\
 & \underline{62.011} \\
965.7000) & 1163.6307\ (1.2+ \text{ Factor of evaporation.} \\
 & \underline{965\ 7000} \\
 & 197\ 93070 \\
 & \underline{193\ 14000}
\end{array}$$

It must be borne in mind that the factor of evaporation varies with the temperature of the steam or temperature of the feed water; therefore, in order to reduce evaporative tests to the standard of from and at 212° Fahrenheit, the factor of evaporation must be obtained from the total heat units in the steam and the feed water, as shown during the test. Then when the factor of evaporation has been determined, the number of pounds of water evaporated per pound of combustible matter in the coal, multiplied by the factor of evaporation, will give the number of pounds of water per pound of combustible matter that would have been evaporated had the feed water been injected into the boiler at a temperature of 212° Fahrenheit, and evaporated at that temperature. It will be observed that before the factor of evaporation can be determined, the total heat units in the steam per pound must first be ascertained, and that can readily be done by reference to the table of saturated steam.

Take the average steam pressure per square inch, registered by the steam gauge during the test, and find in the table the total number of heat units in steam due that pressure, and subtract the heat units in one pound of water due the average temperature of the feed water from the total heat units in the steam per pound due the pressure under which the test was made, and divide the remainder by the constant 965.7, the quotient will give the factor of evaporation according to the rule previously given.

EVAPORATION FROM AND AT 212° FAHRENHEIT.

TO DETERMINE THE EQUIVALENT EVAPORATION OF WATER TO STEAM, FROM AND AT A TEMPERATURE OF 212° FAHRENHEIT.

RULE.—Subtract the heat units in one pound of water due the average temperature of the feed water injected into the boiler during the test, from the total heat units in one pound of steam due the average pressure per square inch shown by the steam gauge during the test, then divide the remainder by the constant 965.7, *and the quotient will give the factor of evaporation. Next multiply the factor of evaporation by the number of pounds of water evaporated per pound of combustible matter in the coal, and the product will give the number of pounds of water per pound of combustible matter, that would have been evaporated had the feed water been injected into the boiler at a temperature of* 212° *Fahrenheit, and evaporated at that temperature.*

Example.—Let 1225.6417 equal total heat units (as shown by table) in one pound of steam at 366.232° Fahrenheit, or at 151.304 pounds gauge pressure.

Let 62.011 equal heat units in feed water at 62° Fahrenheit.

Let 965.7 equal heat units required to evaporate one pound of water from and at 212° Fahrenheit.

Let 7 pounds equal weight of water evaporated per pound of combustible matter in the coal.

Then we have:

$$\left(\frac{1225.6417-62.011}{965.7}\right)\times 7 = 8.4+$$

Pounds of water per pound of combustible that would have been evaporated at a temperature of 212° from feed water at 212° Fahrenheit.

Performing the operation, we have:

$$\begin{array}{rrl} & 1225.6417 & \text{Heat units in steam.} \\ & 62.011 & \text{Heat units in feed water.} \\ \hline 965.7000\,) & 1163.6307\,(\,1.2+ & \text{Factor of evaporation.} \\ & 965.7000 & \\ \hline & 197\,93070 & \\ & 193\,14000 & \\ \hline \end{array}$$

Then, multiplying the factor of evaporation by the number of pounds of water evaporated per pound of combustible matter in the coal, we have:

$$\begin{array}{rl} 1.2 & \\ 7 & \\ \hline 8.4 & \end{array}$$

Pounds of water that would have been evaporated per pound of combustible if the feed water had had a temperature of 212° Fahrenheit and had been evaporated at that temperature.

MOISTURE AND COMBUSTIBLE MATTER IN COAL—HOW DETERMINED.

When evaporative tests are made it is usual to base the calculations upon the combustible matter in the coal, in order to determine the efficiency of the boiler. Ordinarily this is done by placing in a bag exactly 100 pounds of coal of average moisture, taken from different parts of the pile from which the coal for making the evaporative test is to be taken. The weight of the coal must be exactly 100 pounds, exclusive of the weight of the bag in which it is placed. The bag is then tied up and placed in a suitable place during the test to thoroughly dry the coal. After completing the test, the bag of coal is weighed again, and the number of pounds the coal weighs less than 100 represents the percentage of moisture in the coal. For example, if it is one pound, that is one per cent.; if it is two pounds, that is two per cent.; and so on. If it is one per cent., one per cent. is deducted from the total weight of the coal used during the test; if it is two per cent., two per cent. is deducted from the total weight of the coal used; and so on. After deducting the moisture, the weight of the ashes and clinkers is then deducted from the remainder, and that which then remains is termed combustible matter.

The total number of pounds of water evaporated is then divided by the number of pounds of combustible matter in the coal consumed during the trial, and the quotient is taken as the number of pounds of water evaporated per pound of combustible matter.

By dividing the total number of pounds of water evaporated by the total number of pounds of coal consumed during the trial, the quotient will give the number of pounds of water evaporated per pound of coal.

Then, as the feed water is injected into boilers at various temperatures, and as the combustible matter in different coals varies materially, evaporative tests are usually reduced to the standard evaporation, as from and at 212° Fahrenheit. That is supposing the feed water to have been injected into the boiler at 212° Fahrenheit, and evaporated at that temperature. As a further illustration, let it be supposed that the evaporative test was made under an average pressure per square inch of 151.304 pounds, as shown by the steam gauge. By reference to the table, it will be found that the total heat units in one pound of steam at that pressure is 1225.6417, which means that it required 1225.6417 heat units to raise one pound of water from zero to 151.304 pounds pressure per square inch and evaporate it into steam at that pressure. Now, let it be supposed that the feed water had a temperature of 62° Fahrenheit, and to be absolutely correct, by reference to the table of properties of water it will be found that water at a temperature of 62° Fahrenheit contains 62.011 heat units; therefore, the number of heat

units required to evaporate water from a temperature of 62° Fahrenheit would be 62.011 less than it would be from zero. Hence, as the total heat units required to evaporate one pound of water from zero at a pressure by gauge of 151.304 pounds per square inch is 1225.6417, the heat units required to evaporate one pound of water under the same pressure from a temperature of 62° Fahrenheit would be 1225.6417 —62.011=1163.6307 heat units. Now, as it requires 1178.6 heat units to evaporate one pound of water from zero, at 212° Fahrenheit, and as it requires 212.9 heat units to raise water from zero to 212° Fahrenheit, it requires 212.9 heat units less to evaporate it to steam from and at 212° Fahrenheit than it does from zero. Hence, 1178.6—212.9=965.7 heat units required to evaporate one pound of water from and at a temperature of 212° Fahrenheit.

Therefore, as it requires 1163.6307 heat units to evaporate one pound of feed water injected into a boiler at 62° Fahrenheit, under a pressure of 151.304 pounds by gauge, we multiply the number of heat units, 1163.6307, by the number of pounds of water actually evaporated per pound of combustible, and divide the product by 965.7, the number of heat units required to evaporate one pound of water from a temperature of 212° Fahrenheit to steam at that temperature, and the quotient will give the number of pounds of water that would have been evaporated had the feed water been injected into the boiler at 212° Fahrenheit and evaporated at that temperature.

Example.—Let 1163.6307 equal heat units required to evaporate one pound of water from 62° Fahrenheit.

Let 7 pounds equal weight of water actually evaporated per pound of combustible.

Let 965.7 equal number of heat units required to evaporate one pound of water from and at 212° Fahrenheit.

Then we have:

$$\frac{1163.6307 \times 7}{965.7} = 8.4+$$

Pounds of water per pound of combustible that would have been evaporated from and at a temperature of 212° Fahrenheit.

Performing the operation, we have:

```
          1163.6307
                  7
          ---------
965.7000) 8145.4149 (8.4+
          7725 6000
          ---------
           419 81490
           386 28000
           ---------
```

Pounds of water per pound of combustible that would have been evaporated from and at a temperature of 212° Fahrenheit.

Now that the subject has been fully explained, the rule can be given in a brief and simplified form.

REDUCTION OF TESTS TO STANDARD EVAPORATION OF FROM AND AT 212° FAHRENHEIT.

TO REDUCE THE EVAPORATIVE TESTS TO THE STANDARD EVAPORATION OF FROM AND AT 212° FAHRENHEIT.

RULE.—Subtract the heat units in the feed water due its temperature from the total heat units in the steam due the temperature of the steam, then divide the remainder by 965.7, and multiply the quotient by the actual number of pounds of water evaporated per pound of combustible matter in the coal, and the product will give the evaporation as from and at 212° Fahrenheit.

Taking feed water at an average temperature of 62° Fahrenheit, by reference to the table of properties of water, it will be found to contain 62.011 heat units per pound of water. Taking the average pressure of the steam during the test at 151.304 pounds per square inch by the steam gauge, by reference to the table of properties of saturated steam, it will be found that the temperature due that pressure is 366.232° Fahrenheit, and that one pound of steam at that temperature contains a total of 1225.6417 heat units, as shown by the table.

Example.—Let 1225.6417 equal total heat units in steam at a pressure of 151.304 pounds per square inch.
Let 62.011 equal heat units in feed water at a temperature of 62° Fahrenheit
Let 965.7 equal heat units required to evaporate one pound of water from and at 212° Fahrenheit.
Let 7 pounds equal weight of water actually evaporated per pound of combustible matter in the coal.

Then we have:

$$\left(\frac{1225.6417-62.011}{965.7}\right)\times 7 = 8.4+ \quad \text{Pounds of water evaporated as from and at } 212^\circ \text{ Fahrenheit.}$$

Performing the operation, we have:

```
            1225.6417
              62.011
            ---------
965.7000) 1163.6307  ( 1.2+
           965 7000     7
           --------   ----
           197 93070  8.4+  Pounds of water evaporated
           193 14000          as from and at 212° Fahrenheit.
           ---------
```

NOTE.—Heat units are usually reckoned from 32° Fahrenheit, but for greater simplicity they have been taken at zero Fahrenheit, upon which the foregoing calculations are based. The tables of properties of saturated steam and properties of water are arranged upon the same basis.

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure per Square Inch.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
1	....	102.	102.086	1042.964	1145.050	.0030	20620.0
2	....	126.266	126.440	1026.010	1152.450	.0058	10720.0
3	....	141.622	141.877	1015.254	1157.131	.0085	7326.0
4	......	153.070	153.396	1007.229	1160.625	.0112	5600.0
5	....	162.330	162.722	1000.727	1163.449	.0137	4535.0
6	....	170.123	170.577	995.249	1165.826	.0163	3814.0
7	....	176.910	177.425	990.471	1167.896	.0189	3300.0
8	....	182.910	183.481	986.245	1169.726	.0214	2910.0
9	......	188.316	188.941	982.434	1171.375	.0239	2607.0
10	... ...	193.240	193.919	978.958	1172.877	.0264	2360.0
11	....	197.768	198.496	975.762	1174.258	.0289	2157.0
12	....	201.960	202.737	972.800	1175.537	.0313	1988.0
13	....	205.885	206.709	970.025	1176.734	.0337	1846.0
14	....	209.560	210.428	967.427	1177.855	.0362	1722.0
14.7	....	212.000	212.900	965.700	1178.600	.03797	1644.0
15	.304	213.025	213.939	964.973	1178.912	.0387	1612.0
16	1.304	216.296	217.252	962.657	1179.909	.0413	1514.0
17	2.304	219.410	220.409	960.450	1180.859	.0437	1427.0
18	3.304	222.378	223.419	958.345	1181.764	.0462	1350.6
19	4.304	225.203	226.285	956.343	1182.628	.0487	1282.1
20	5.304	227.917	229.039	954.415	1183.454	.0511	1220.3
21	6.304	230.515	231.676	952.570	1184.246	.0536	1164.4
22	7.304	233.017	234.218	950.791	1185.009	.0561	1113.5
23	8.304	235.432	236.672	949.072	1185.744	.0585	1066.9
24	9.304	237.752	239.029	947.424	1186.453	.0610	1024.1
25	10.304	240.000	241.314	945.825	1187.139	.0634	984.8
26	11.304	242.175	243.526	944.277	1187.803	.0658	948.4
27	12.304	244.284	245.671	942.775	1188.446	.0683	914.6
28	13.304	246.326	247.748	941.321	1189.069	.0707	883.2
29	14.304	248.310	249.769	939.905	1189.674	.0731	854.0
30	15.304	250.245	251.738	938.925	1190.263	.0755	826.8
31	16.304	252.122	253.648	937.1878	1190.8358	.0779	801.2
32	17.304	253.952	255.512	935.8818	1191.3938	.0803	777.2
33	18.304	255.735	257.329	934.6088	1191.9378	.0827	754.7
34	19.304	257.476	259.103	933.3658	1192.4688	.0851	733.5
35	20.304	259.176	260.835	932.1523	1192.9873	.0875	713.4
36	21.304	260.835	262.527	930.9668	1193.4938	.0899	694.5
37	22.304	262.458	264.182	929.8068	1193.9888	.0922	676.6
38	23.304	264.045	265.801	928.6718	1194.4728	.0946	659.7
39	24.304	265.599	267.386	927.5608	1194.9468	.0970	643.6
40	25.304	267.120	268.938	926.4728	1195.4108	.0994	628.2
41	26.304	268.611	270.460	925.4058	1195.8658	.1017	613.4
42	27.304	270.073	271.954	924.3578	1196.3118	.1041	599.3
43	28.304	271.507	273.417	923.3323	1196.7493	.1064	586.1
44	29.304	272.915	274.855	922.3238	1197.1788	.1088	573.7
45	30.304	274.296	276.266	921.3343	1197.6003	.1111	561.8
46	31.304	275.652	277.651	920.3632	1198.0142	.1134	550.4
47	32.304	276.986	279.016	919.4052	1198.4212	.1158	539.5
48	33.304	278.297	280.355	918.4662	1198.8212	.1181	529.0
49	34.304	279.585	281.672	917.5422	1199.2142	.1204	518.6
50	35.304	280.854	282.969	916.6316	1199.6006	.1227	508.5
51	36.304	282.099	284.243	915.7377	1199.9807	.1251	499.1
52	37.304	283.326	285.499	914.8557	1200.3547	.1274	490.1

TABLE OF PROPERTIES OF SATURATED STEAM—Continued.

Pressure per Square Inch. Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.	Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit). Number of Units of Heat in Water	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.	Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
53	38.304	284.534	286.736	913.9871	1200.7231	.1297	481.4
54	39.304	285.724	287.952	913.1340	1201.0860	.1320	472.9
55	40.304	286.897	289.153	912.2906	1201.4436	.1343	464.7
56	41.304	288.052	290.335	911.4611	1201.7961	.1366	457.0
57	42.304	289.112	291.503	910.6407	1202.1437	.1388	449.6
58	43.304	290.316	292.654	909.8325	1202.4865	.1411	442.4
59	44.304	291.425	293.790	909.0346	1202.8246	.1434	435.3
60	45.304	292.520	294.911	908.2472	1203.1582	.1457	428.5
61	46.304	293.598	296.016	907.4713	1203.4873	.14793	422.0
62	47.304	294.663	297.108	906.7042	1203.8122	.15021	415.6
63	48.304	295.714	298.185	905.9477	1204.1329	.15248	409.4
64	49.304	296.752	299.249	905.2005	1204.4495	.15471	403.5
65	50.304	297.777	300.300	904.4621	1204.7621	.15697	397.7
66	51.304	298.789	301.338	903.7327	1205.0707	.15921	392.1
67	52.304	299.789	302.364	903.0116	1205.3756	.16147	386.6
68	53.304	300.776	303.377	902.2999	1205.6769	.16372	381.3
69	54.304	301.753	304.380	901.5947	1205.9747	.16598	376.1
70	55.304	302.718	305.370	900.8991	1206.2691	.16817	371.2
71	56.304	303.673	306.350	900.2101	1206.5601	.17038	366.4
72	57.304	304.617	307.320	899.5280	1206.8480	.17259	361.7
73	58.304	305.551	308.279	898.8537	1207.1327	.17481	357.1
74	59.304	306.474	309.228	898.1863	1207.4143	.17704	352.6
75	60.304	307.388	310.166	897.5269	1207.6929	.17923	348.3
76	61.304	308.290	311.092	896.8764	1207.9684	.18142	344.1
77	62.304	309.184	312.011	896.2301	1208.2411	.18360	340.0
78	63.304	310.069	312.920	895.5910	1208.5110	.18579	336.0
79	64.304	310.945	313.821	894.9571	1208.7781	.18797	332.1
80	65.304	311.812	314.712	894.3304	1209.0424	.19015	328.3
81	66.304	312.670	315.595	893.7092	1209.3042	.19232	324.6
82	67.304	313.520	316.468	893.0954	1209.5634	.19454	320.9
83	68.304	314.361	317.333	892.4871	1209.8201	.19674	317.3
84	69.304	315.195	318.190	891.8843	1210.0743	.19887	313.9
85	70.304	316.021	319.040	891.2862	1210.3262	.20105	310.5
86	71.304	316.839	319.882	890.6938	1210.5758	.20321	307.2
87	72.304	317.650	320.717	890.1061	1210.8231	.20535	304.0
88	73.304	318.453	321.543	889.5251	1211.0681	.20753	300.8
89	74.304	319.249	322.362	888.9490	1211.3110	.20970	297.7
90	75.304	320.039	323.176	888.3758	1211.5518	.21183	294.7
91	76.304	320.821	323.981	887.8094	1211.7904	.21393	291.8
92	77.304	321.597	324.781	887.2460	1212.0270	.21608	288.9
93	78.304	322.366	325.572	886.6896	1212.2616	.21829	286.1
94	79.304	323.128	326.358	886.1362	1212.4942	.22045	283.3
95	80.304	323.884	327.136	885.5887	1212.7247	.22247	280.6
96	81.304	324.634	327.909	885.0444	1212.9534	.22455	278.0
97	82.304	325.378	328.675	884.5052	1213.1802	.22667	275.4
98	83.304	326.114	329.433	883.9721	1213.4051	.22883	272.8
99	84.304	326.845	330.186	883.4421	1213.6281	.23095	270.3
100	85.304	327.571	330.935	882.9144	1213.8494	.23302	267.9
101	86.304	328.291	331.678	882.3909	1214.0689	.23510	265.5
102	87.304	329.005	332.414	881.8727	1214.2867	.23717	263.2
103	88.304	329.714	333.145	881.3577	1214.5027	.23925	260.9
104	89.304	330.416	333.869	880.8481	1214.7171	.24132	258.7
105	90.304	331.113	334.587	880.3429	1214.9299	.24340	256.5

TABLE OF PROPERTIES OF SATURATED STEAM—Continued.

Pressure per Square Inch.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
106	91.304	331.805	335.301	879.8400	1215.1410	.24547	254.3
107	92.304	332.492	336.009	879.3416	1215.3506	.24754	252.2
108	93.304	333.174	336.714	878.8447	1215.5587	.24961	250.1
109	94.304	333.851	337.411	878.3542	1215.7652	.25168	248.0
110	95.304	334.523	338.105	877.8653	1215.9703	.25376	246.0
111	96.304	335.191	338.795	877.3789	1216.1739	.25582	244.0
112	97.304	335.854	339.479	876.8970	1216.3760	.25788	242.0
113	98.304	336.511	340.157	876.4198	1216.5768	.25994	240.1
114	99.304	337.165	340.832	875.9442	1216.7762	.26199	238.2
115	100.304	337.814	341.502	875.4721	1216.9741	.26405	236.3
116	101.304	338.459	342.169	875.0018	1217.1708	.26611	234.5
117	102.304	339.100	342.831	874.5352	1217.3662	.26816	232.7
118	103.304	339.736	343.488	874.0722	1217.5602	.27020	231.0
119	104.304	340.368	344.141	873.6120	1217.7530	.27224	229.3
120	105.304	340.995	344.789	873.1555	1217.9445	.27428	227.6
121	106.304	341.618	345.432	872.7027	1218.1347	.27628	226.0
122	107.304	342:238	346.073	872.2508	1218.3238	.27828	224.4
123	108.304	342.854	346.709	871.8027	1218.5117	.28027	222.8
124	109.304	343.466	347.343	871.3553	1218.6983	.28227	221.2
125	110.304	344.074	347.972	870.9118	1218.8838	.28426	219.7
126	111.304	344.678	348.596	870.4721	1219.0681	.28625	218.2
127	112.304	345.279	349.217	870.0342	1219.2512	.28824	216.7
128	113.304	345.876	349.835	869.5983	1219.4333	.29023	215.2
129	114.304	346.459	350.448	869.1663	1219.6143	.29222	213.7
130	115.304	347.059	351.059	868.7351	1219.7941	.29420	212.3
131	116.304	347.644	351.665	868.3079	1219.9729	.29618	210.9
132	117.304	348.227	352.267	867.8836	1220.1506	.29816	209.5
133	118.304	348.806	352.867	867.4601	1220.3271	.30013	208.1
134	119.304	349.382	353.463	867.0397	1220.5027	.30209	206.7
135	120.304	349.954	354.055	866.6223	1220.6773	.30405	205.4
136	121.304	350.523	354.644	866.2068	1220.8508	.30601	204.1
137	122.304	351.089	355.230	866.7934	1221.0234	.30796	202.8
138	123.304	351.752	355.813	865.3820	1221.1950	.30990	201.5
139	124.304	352.211	356.392	864.9735	1221.3655	.31186	200.2
140	125.304	352.767	356.969	864.5661	1221.5351	.31386	199.0
141	126.304	353.319	357.541	864.1627	1221.7037	.31587	197.8
142	127.304	353.869	358.110	863.7613	1221.8713	.31788	196.6
143	128.304	354.416	358.677	863.3611	1222.0381	.31990	195.4
144	129.304	354.960	359.240	862.9640	1222.2040	.32190	194.2
145	130.304	355.501	359.801	862.5679	1222.3689	.32391	193.0
146	131.304	356.039	360.359	862.1740	1222.5330	.32592	191.9
147	132.304	356.574	360.913	861.7832	1222.6962	.32794	190.8
148	133.304	357.106	361.465	861.3934	1222.8584	.32995	189.7
149	134.304	357.635	362.013	861.0068	1223.0198	.33196	188.6
150	135.304	358.161	362.559	860.6213	1223.1803	.33400	187.5
151	136.304	358.683	363.100	860.2399	1223.3399	.33580	186.4
152	137.304	359.203	363.640	859.8588	1223.4988	.33761	185.3
153	138.304	359.721	364.177	859.4799	1223.6569	.33942	184.3
154	139.304	360 236	364.711	859.1031	1223.8141	.34123	183.3
155	140.304	360.749	365.243	858.7276	1223.9706	.34304	182.3
156	141.304	361.260	365.773	858.3533	1224.1263	.34485	181.3
157	142.304	361.768	366.300	857.9811	1224.2811	.34666	180.3
158	143.304	362.273	366.824	857.6112	1224.4352	.34847	179.3

TABLE OF PROPERTIES OF SATURATED STEAM—Continued.

PRESSURE PER SQUARE INCH.		Temperature in Fahrenheit Degrees.	NUMBER OF BRITISH THERMAL UNITS IN ONE POUND FROM ZERO (FAHRENHEIT).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
159	144.304	362.776	367.347	857.2415	1224.5885	.35028	178.3
160	145.304	363.277	367.867	856.8740	1224.7410	.35209	177.3
161	146.304	363.774	368.383	856.5099	1224.8929	.35397	176.4
162	147.304	364.270	368.898	856.1461	1225.0441	.35585	175.5
163	148.304	364.764	369.410	855.7846	1225.1946	.35773	174.6
164	149.304	365.255	369.920	855.4243	1225.3443	.35961	173.7
165	150.304	365.744	370.428	855.0654	1225.4934	.36149	172.8
166	151.304	366.232	370.934	854.7077	1225.6417	.36337	171.9
167	152.304	366.717	371.438	854.3514	1225.7894	.36525	171.0
168	153.304	367.199	371.939	853.9974	1225.9364	.36714	170.1
169	154.304	367.680	372.437	853.6456	1226.0826	.36903	169.2
170	155.304	368.158	372.934	853.2942	1226.2282	.37092	168.4
171	156.304	368.632	373.427	852.9461	1226.3731	.37272	167.6
172	157.304	369.105	373.918	852.5995	1226.5175	.37452	166.8
173	158.304	369.576	374.408	852.2533	1226.6613	.37632	166.0
174	159.304	370.045	374.895	851.9094	1226.8044	.37812	165.2
175	160.304	370.512	375.380	851.5670	1226.9470	.37992	164.4
176	161.304	370.978	375.865	851.2239	1227.0889	.38172	163.6
177	162.304	371.442	376.347	850.8833	1227.2303	.38353	162.8
178	163.304	371.904	376.827	850.5441	1227.3711	.38534	162.0
179	164.304	372.364	377.305	850.2062	1227.5112	.38715	161.2
180	165.304	372.822	377.781	849.8698	1227.6508	.38895	160.4
181	166.304	373.275	378.255	849.5347	1227.7897	.39077	159.7
182	167.304	373.731	378.727	849.2011	1227.9281	.39259	159.0
183	168.304	374.183	379.197	848.8689	1228.0659	.39441	158.3
184	169.304	374.633	279.665	848.5380	1228.2030	.39624	157.6
185	170.304	375.081	380.131	848.2086	1228.3396	.39807	156.9
186	171.304	375.527	380.595	847.8805	1228.4755	.39990	156.2
187	172.304	375.971	381.056	847.5549	1228.6109	.40173	155.5
188	173.304	376.413	381.516	847.2297	1228.7457	.40356	154.8
189	174.304	376.853	381.974	846.9058	1228.8798	.40539	154.1
190	175.304	377.291	382.429	846.5844	1229.0134	.40722	153.4
191	176.304	377.727	382.883	846.2633	1229.1463	.40899	152.7
192	177.304	378.161	383.335	845.9437	1229.2787	.41076	152.0
193	178.304	378.593	383.785	845.6256	1229.4106	.41253	151.3
194	179.304	379.023	384.233	845.3089	1229.5419	.41430	150.7
195	180.304	379.452	384.679	844.9938	1229.6728	.41607	150.1
196	181.304	379.979	385.123	844.6801	1229.8031	.41784	149.5
197	182.304	380.305	385.567	844.3660	1229.9330	.41962	148.9
198	183.304	380.729	386.008	844.0543	1230.0623	.42140	148.3
199	184.304	381.152	386.449	843.7422	1230.1912	.42318	147.7
200	185.304	381.573	386.887	843.4326	1230.3196	.42496	147.1
201	186.304	381.992	387.324	843.1234	1230.4474	.42667	146.5
202	187.304	382.410	387.760	842.8148	1230.5748	.42838	145.9
203	188.304	382.827	388.194	842.5076	1230.7016	.43009	145.3
204	189.304	383.242	388.627	842.2010	1230.8280	.43180	144.7
205	190.304	383.655	389.057	841.8969	1230.9539	.43351	144.1
206	191.304	384.066	389.485	841.5942	1231.0792	.43523	143.5
207	192.304	384.475	389.912	841.2921	1231.2041	.43695	142.9
208	193.304	384.883	390.337	840.9914	1231.3284	.43866	142.3
209	194.304	385.288	390.759	840.6933	1231.4523	.44039	141.8
210	195.304	385.671	391.179	840.3967	1231.5757	.44211	141.3

CHAPTER XII.

CALORIMETER TESTS.

There is, perhaps, no operation connected with the science of steam engineering requiring more care and accuracy than that of making calorimeter tests.

The rules and formulæ laid down in many excellent works on steam engineering are commendable for their accuracy, and in some cases, simplicity; yet few, if any, are calculated to reach the comprehension or understanding of the student, unless he is proficient in the higher branches of mathematics. It will therefore be the purpose here to present this important subject in its utmost simplicity, in order that engineers of limited education may get the benefit of the science embraced therein.

Evaporative tests, without the calorimeter tests, really prove but little, so far as the efficiency of a steam boiler is concerned. To determine the real efficiency of a steam boiler the quality of the steam generated must be ascertained, as well as the quantity of water evaporated per pound of coal, or per pound of combustible. Without the calorimeter test the faulty constructed boiler may, if judged simply from the amount of water evaporated, show greater efficiency than the properly constructed boiler; because the imperfect or faulty boiler is liable to carry off more water with the steam than the properly constructed boiler. Therefore, every pound of surplus water carried off with the steam, robs the boiler of the heat due the quantity of surplus water carried off with the steam without producing an equivalent in work performed. On the contrary, it may result largely detrimental to the performance of the engine supplied with steam from such a boiler.

The main object then, is to get steam as dry as possible without superheating it, and the boiler that will furnish the driest steam and perform the greatest amount of work in proportion to the amount of fuel consumed is the best boiler, all other things being equal, such as proper setting of boilers, properly constructed furnaces and proper proportions of smoke stack, etc.

One of the simplest devices for making calorimeter tests is that shown in Fig. 124.

A represents the main steam pipe, and shows how the attachment is made. B represents a post for holding up the calorimeter pipe. The

portion of the pipe inside of the main steam pipe should be perforated with $\frac{1}{8}$ inch holes, and the pipe leading to the calorimeter or barrel should be made of $\frac{1}{2}$ inch gas pipe. The valve attached to the end of that pipe should have a petcock screwed into the top for the purpose

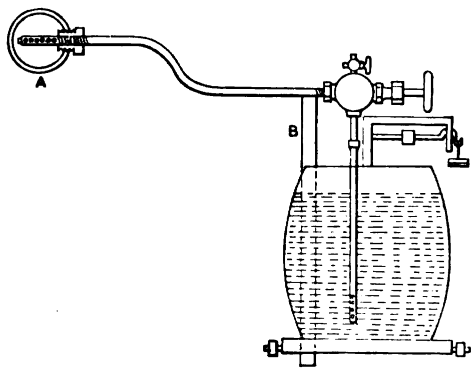


Fig. 124.

of opening it to allow any water in the pipe leading into the barrel to fall to the level of the water in the barrel, when the test has been completed.

The barrel employed for this purpose should be in condition to absorb as little water as possible while making the test. The down pipe from the valve should have a small rubber hose attached to the end of the pipe, as shown in the diagram, and reach to within a short distance of the bottom of the barrel. The lower end of the hose should be closed, and the hose above that perforated all around with small holes.

Before the water is put into the barrel, the barrel should be accurately weighed and the weight carefully noted, after which put an even number of pounds of water into the barrel for convenient calculation, and leave sufficient room for the desired amount of condensed steam. Fractions of pounds should be avoided as much as possible.

Next, set the scales so that they will balance when five or six per cent. more water has been added in the form of condensed steam.

Next, take the hose out of the barrel, open the valve, and let the steam blow through so as to heat the pipes thoroughly. While the steam is blowing through, take the temperature of the water in the barrel and carefully note the same. Shut the steam off and then insert the hose into the barrel, then turn the steam on, and in a short time place the hand under the beam of the scales, to ascertain by lifting, if it is about to balance, and as soon as it is found that the scales are about to balance, shut the steam off and open the petcock and balance the scales. Then place the thermometer in the water, stir it around

and carefully observe the highest temperature reached. Note the same, and also note the weight of water in the barrel.

We will now suppose the barrel to have contained 360 pounds of water before adding the condensed steam, and that the temperature of the water was 50° Fahrenheit. By reference to the table, it will be found that water at a temperature of 50° Fahrenheit contains 50.003 heat units per pound of water.

Then we have:

360 pounds equal weight of water in the barrel before adding steam.
50.003 equal number of heat units per pound of water at a temperature of 50° Fahrenheit.

Next, suppose the steam to have raised the temperature of the water in the barrel to 110° Fahrenheit. By referring to the table, it will be found that water, at a temperature of 110° Fahrenheit, contains 110.110 heat units per pound of water.

Then we have:

360 pounds equal weight of water in the barrel before adding steam.
50.003 equal number of heat units per pound of water at a temperature of 50° Fahrenheit, before adding steam.
110.110 equal number of heat units per pound of water at 110° Fahrenheit, the temperature after adding steam.

Next, suppose that 20 pounds of steam and water have been added to the water in the barrel from the steam pipe, and we will have 380 pounds of water in the barrel.

Then we have:

360 pounds equal weight of water in the barrel before adding steam.
50.003 equal number of heat units per pound of water at a temperature of 50° Fahrenheit, before adding steam.
110.110 equal number of heat units per pound of water at 110° Fahrenheit, the temperature after adding steam.
20 pounds equal weight of condensed steam and water added to water in the barrel.
380 pounds equal weight of water in the barrel after adding condensed steam and water.

Next, suppose the steam pressure per square inch indicated by the steam gauge during the trial to have been 100.304 pounds, by reference to the table, it will be found that the total heat per pound of dry steam at that pressure is 1216.9741 units.

Then we have:

360 pounds equal weight of water in the barrel before adding steam.

50.003 equal number of heat units per pound of water at 50° Fahrenheit before adding steam.

110.110 equal number of heat units per pound of water at 110° Fahrenheit, the temperature after adding steam.

20 pounds equal weight of condensed steam and water added to water in the barrel.

380 pounds equal weight of water in the barrel after adding condensed steam and water.

1216.9741 equal number of heat units in one pound of dry steam at 100.304 pounds gauge pressure.

PERCENTAGE OF WATER IN STEAM.

TO DETERMINE THE PERCENTAGE OF WATER IN STEAM.

RULE.—First, subtract the number of pounds of water contained in the barrel before the condensed steam was added from the total number of pounds in the barrel after the condensed steam was added, and multiply the remainder by the total heat units, as shown by the table, contained in one pound of steam due the pressure per square inch indicated by the steam gauge during the time the test was made, and the product will give the total heat units that would have been contained in the steam that has been discharged in the barrel if the steam had been dry, and call the product "Product No. 1."

Second, multiply the number of heat units contained in one pound of the heated water in the barrel by the number of pounds of such water, and the product will give the total number of heat units contained in the heated water in the barrel, and call the product "Product No. 2."

Third, multiply the number of heat units contained in one pound of the unheated water in the barrel by the number of pounds of that water, and the product will give the total number of heat units in the unheated water in the barrel, and call the product, "Product No. 3."

Fourth, subtract the total heat units in the unheated water in the barrel from the total heat units contained in the water after being heated; in other words, subtract "Product No. 3" from "Product No. 2" and the remainder will give the number of heat units that have been added by the steam and water discharged into the barrel from the steam pipe, and call the remainder "Remainder No. 1."

Fifth, subtract "Remainder No. 1" (the heat units that have been added to the original water in the barrel) from "Product No. 1" (the total heat units that would have been contained in the steam if the steam had been dry), and the remainder will show the difference in heat units between dry steam and the steam discharged in the barrel and call this "Remainder No. 2."

Sixth, multiply the total heat units, as shown in the table, contained in one pound of steam due the pressure per square inch as shown by the gauge during the test, by the number of pounds of steam and water that have been added to the original water in the barrel, and the product will give the total number of heat units that would have been contained in the steam that has been discharged in the barrel if the steam had been dry, and call this product "Product No. 4."

Seventh, divide "Remainder No. 2" (the difference in heat units between dry steam and the steam discharged in the barrel) by "Product No. 4," and multiply the quotient by 100, and the product will give the per cent. of water contained in the steam discharged into the barrel.

Example.—Let 1216.9741 equal heat units in one pound of dry steam under 100.304 pounds gauge pressure.

Let 380 pounds equal weight of water in the barrel after adding steam.

Let 360 pounds equal weight of water in the barrel before adding steam.

Let 110.110 equal heat units contained per pound of water after adding steam, temperature being 110° Fahrenheit.

Let 50.003 equal heat units contained per pound of original water in the barrel at a temperature of 50° Fahrenheit.

Let 20 pounds equal weight of steam and water added to original water in the barrel.

Then we have:

$$\left(\frac{(1216.9741\times\overline{380-360})-(110.110\times 380-\overline{50.003\times 360})}{1216.9741\times 20}\right)\times 100=2.04+$$

Per cent. of water in steam discharged in the barrel.

Performing the operation, we have:

380	Pounds of water in barrel after adding steam.
360	Pounds of water in barrel before adding steam.
20	Pounds of water added by the steam to the water in the barrel.
1216.9741	Heat units contained in one pound of dry steam at 100.304 pounds pressure
24339.4820	"Product No. 1." Total heat units that would have been contained in the steam discharged in the barrel if the steam had been dry.

Next we have:

110.110	Heat units per pound of water in barrel after adding steam.
380	Pounds of water in barrel after adding steam.
8808 800	
33033 0	
41841.800	"Product No. 2." Total number of heat units in 380 pounds of water at a temperature of 110° Fahrenheit.

Next we have:

50.003	Heat units per pound of water before adding steam.
360	Pounds of water in barrel before adding steam.
3000 180	
15000 9	
18001.080	"Product No. 3." Total number of heat units in 360 pounds of water at a temperature of 50° Fahrenheit.

Next we have:

41841.800	"Product No. 2."
18001.080	"Product No. 3."
23840.720	"Remainder No. 1." The number of heat units which have been added to the water in the barrel by the steam and water discharged into it.

Next we have:

24339.482	"Product No. 1."
23840.720	"Remainder No. 1."
498.762	"Remainder No. 2." Heat units, difference between dry steam and the steam discharged into the barrel.

Next we have:

1216.9741	Heat units in one pound of dry steam at 100.304 pounds pressure.
20	Pounds of steam and water added to the original water in barrel.
24339.4820	"Product No. 4." Total heat units that would have been contained in the 20 pounds of steam discharged into the barrel if the steam had been dry.

Finally, we have:

24339.482) 498.76200 (0.0204+
486 78964   100

11 9723600   2.0400 Per cent. of water in the steam discharged into the barrel.
9 7357928

If the steam is superheated it will show a greater number of heat units per pound for a given pressure than is contained in the standard steam as shown in the table.

TABLE OF PROPERTIES OF WATER FROM 32° TO 212° FAHRENHEIT.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
.089	.1811	32	32.000	1091.700	1123.700	.00030	208,080
.092	.1884	33	33.000	1091.005	1124.005	.00030	200,480
.096	.1960	34	34.000	1090.310	1124.310	.00031	193,180
.100	.2039	35	35.000	1089.615	1124.615	.00032	186,180
.104	.2121	36	36.000	1088.920	1124.920	.00033	179,380
.108	.2205	37	37.000	1088.225	1125.225	.00034	172,780
.112	.2292	38	38.000	1087.530	1125.530	.00036	166,380
.117	.2382	39	39.001	1086.834	1125.835	.00038	160,230
.122	.2476	40	40.001	1086.139	1126.140	.00040	154,330
.127	.2573	41	41.001	1085.444	1126.445	.00042	148,620
.132	.2673	42	42.001	1084.749	1126.750	.00043	143,220
.137	.2777	43	43.001	1084.054	1127.055	.00045	138,070
.142	.2884	44	44.002	1083.358	1127.360	.00047	133,120
.147	.2994	45	45.002	1082.663	1127.665	.00049	128,370
.152	.3109	46	46.002	1081.968	1127.970	.00050	123,840
.158	.3228	47	47.002	1081.273	1128.275	.00052	119,610
.164	.3351	48	48.003	1080.577	1128.580	.00054	115,490
.170	.3478	49	49.003	1079.882	1128.885	.00056	111,470
.176	.3608	50	50.003	1079.187	1129.190	.00058	107,630
.183	.3743	51	51.004	1078.491	1129.495	.00060	103,930
.190	.3883	52	52.004	1077.796	1129.800	.00062	100,330
.197	.4028	53	53.005	1077.100	1130.105	.00065	96,930
.205	.4177	54	54.005	1076.405	1130.410	.00067	93,680
.212	.4332	55	55.006	1075.709	1130.715	.00069	90,540
.220	.4492	56	56.006	1075.014	1131.020	.00071	87,500
.228	.4656	57	57.007	1074.318	1131.325	.00073	84,560
.236	.4825	58	58.007	1073.623	1131.630	.00076	81,740
.245	.5000	59	59.008	1072.927	1131.935	.00079	79,020
.254	.5180	60	60.009	1072.231	1132.240	.00082	76,370
.263	.5367	61	61.010	1071.535	1132.545	.00085	73,810
.273	.5560	62	62.011	1070.839	1132.850	.00088	71,330
.282	.5758	63	63.012	1070.143	1133.155	.00091	68,940
.292	.5962	64	64.013	1069.447	1133.460	.00094	66,630
.302	.6173	65	65.014	1068.751	1133.765	.00097	64,420
.313	.6391	66	66.015	1068.055	1134.070	.00100	62,290
.324	.6615	67	67.016	1067.359	1134.375	.00103	60,280
.335	.6846	68	68.018	1066.662	1134.680	.00107	58,340
.347	.7084	69	69.019	1065.966	1134.985	.00111	56,470
.359	.7330	70	70.020	1065.270	1135.290	.00115	54,660
.372	.7583	71	71.021	1064.574	1135.595	.00119	52,910
.385	.7844	72	72.023	1063.877	1135.900	.00123	51,210
.398	.8114	73	73.024	1063.181	1136.205	.00127	49,570
.411	.8391	74	74.026	1062.484	1136.510	.00131	48,000
.425	.8676	75	75.027	1061.788	1136.815	.00135	46,510
.440	.8969	76	76.029	1061.091	1137.120	.00139	45,060
.455	.9271	77	77.030	1060.395	1137.425	.00143	43,650
.470	.9583	78	78.032	1059.698	1137.730	.00148	42,280
.486	.9905	79	79.034	1059.001	1138.035	.00153	40,960
.502	1.023	80	80.036	1058.304	1138.340	.00158	39,690
.518	1.056	81	81.037	1057.608	1138.645	.00163	38,480
.535	1.091	82	82.039	1056.911	1138.950	.00168	37,320

Table of
Properties of Water from 32° to 212° Fahrenheit.—Continued.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
.553	1.127	83	83.041	1056.214	1139.255	.00173	36,190
.571	1.163	84	84.043	1055.517	1139.560	.00178	35,100
.590	1.201	85	85.045	1054.820	1139.865	.00183	34,050
.609	1:240	86	86.047	1054.123	1140.170	.00189	33,030
.629	1.281	87	87.049	1053.426	1140.475	.00195	32,050
.650	1.323	88	88.051	1052.729	1140.780	.00201	31,100
.671	1.366	89	89.053	1052.032	1141.085	.00207	30,180
.692	1.410	90	90.055	1051.335	1141.390	.00213	29,290
.715	1.454	91	91.057	1050.638	1141.695	.00219	28,430
.738	1.500	92	92.059	1049.941	1142.000	.00226	27,600
.761	1.548	93	93.061	1049.244	1142.305	.00233	26,800
.785	1.597	94	94.063	1048.547	1142.610	.00240	26,020
.809	1.647	95	95.065	1047.850	1142.915	.00247	25,270
.834	1.698	96	96.068	1047.152	1143.220	.00254	24,540
.860	1.751	97	97.071	1046.454	1143.525	.00262	23,830
.887	1.805	98	98.074	1045.756	1143.830	.00270	23,140
.914	1.861	99	99.077	1045.058	1144.135	.00278	22,470
.943	1.918	100	100.080	1044.360	1144.440	.00286	21,830
.972	1.977	101	101.083	1043.662	1144.745	.00294	21,210
1.001	2.037	102	102.086	1042.964	1145.050	.00302	20,620
1.031	2.099	103	103.089	1042.266	1145.355	.00311	20,050
1.062	2.163	104	104.092	1041.568	1145.660	.00320	19,500
1.094	2.227	105	105.095	1040.870	1145.965	.00330	18,970
1.126	2.293	106	106.098	1040.172	1146.270	.00340	18,460
1.159	2.361	107	107.101	1039.474	1146.575	.00350	17,960
1.193	2.431	108	108.104	1038.776	1146.880	.00360	17,470
1.229	2.503	109	109.107	1038.078	1147.185	.00370	16,990
1.265	2.577	110	110.110	1037.380	1147.490	.00380	16,520
1.302	2.653	111	111.113	1036.682	1147.795	.00390	16,070
1.341	2.731	112	112.117	1035.983	1148.100	.00400	15,640
1.381	2.810	113	113.121	1035.284	1148.405	.00410	15,220
1.421	2.892	114	114.125	1034.585	1148.710	.00421	14,820
1.462	2.976	115	115.129	1033.886	1149.015	.00433	14,430
1.504	3.061	116	116.133	1033.187	1149.320	.00445	14,050
1.547	3.149	117	117.137	1032.488	1149.625	.00457	13,680
1.591	3.239	118	118.141	1031.789	1149.930	.00470	13,320
1.636	3.331	119	119.145	1031.090	1150.235	.00483	12,970
1.682	3.425	120	120.149	1030.391	1150.540	.00496	12,630
1.730	3.522	121	121.153	1029.692	1150.845	.00508	12,300
1.779	3.621	122	122.157	1028.993	1151.150	.00521	11,980
1.828	3.723	123	123.161	1028.294	1151.455	.00535	11,670
1.879	3.826	124	124.165	1027.595	1151.760	.00549	11,370
1.931	3.933	125	125.169	1026.896	1152.065	.00563	11,080
1.984	4.042	126	126.173	1026.197	1152.370	.00578	10,800
2.039	4.153	127	127.177	1025.498	1152.675	.00593	10,530
2.096	4.267	128	128.182	1024.798	1152.980	.00608	10,265
2.154	4.384	129	129.187	1024.098	1153.285	.00624	10,010
2.213	4.503	130	130.192	1023.398	1153.590	.00640	9,760
2.273	4.625	131	131.197	1022.698	1153.895	.00656	9,516
2.335	4.750	132	132.202	1021.998	1154.200	.00673	9,276
2.398	4.878	133	133.207	1021.298	1154.505	.00690	9,046

TABLE OF
PROPERTIES OF WATER FROM 32° TO 212° FAHRENHEIT.—Continued.

ELASTIC FORCE.		Temperature in Fahrenheit Degrees.	NUMBER OF BRITISH THERMAL UNITS IN ONE POUND FROM ZERO (FAHRENHEIT).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
2.461	5.009	134	134.212	1020.598	1154.810	.00707	8,826
2.526	5.143	135	135.217	1019.898	1155.115	.00725	8,611
2.594	5.280	136	136.222	1019.198	1155.420	.00743	8,401
2.663	5.420	137	137.227	1018.498	1155.725	.00761	8,191
2.732	5.563	138	138.223	1017.797	1156.030	.00780	7,991
2.803	5.709	139	139.239	1017.096	1156.335	.00799	7,798
2.876	5.858	140	140.245	1016.395	1156.640	.00819	7,613
2.952	6.011	141	141.251	1015.694	1156.945	.00839	7,433
3.029	6.167	142	142.257	1014.993	1157.250	.00860	7,258
3.108	6.327	143	143.263	1014.292	1157.555	.00881	7,088
3.188	6.490	144	144.269	1013.591	1157.860	.00903	6,920
3.270	6.657	145	145.275	1012.890	1158.165	.00925	6,755
3.353	6.827	146	146.281	1012.189	1158.470	.00948	6,595
3.438	7.001	147	147.287	1011.488	1158.775	.00971	6,440
3.526	7.179	148	148.293	1010.787	1159.080	.00993	6,290
3.615	7.361	149	149.299	1010.086	1159.385	.01016	6,144
3.707	7.547	150	150.305	1009.385	1159.690	.01040	6,004
3.801	7.736	151	151.311	1008.684	1159.995	.01064	5,867
3.896	7.929	152	152.318	1007.982	1160.300	.01089	5,734
3.992	8.127	153	153.325	1007.280	1160.605	.01114	5,604
4.090	8.329	154	154.332	1006.578	1160.910	.01140	5,477
4.191	8.535	155	155.339	1005.876	1161.215	.01167	5,353
4.295	8.745	156	156.346	1005.174	1161.520	.01194	5,232
4.400	8.959	157	157.353	1004.472	1161.825	.01222	5,114
4.507	9.178	158	158.360	1003.770	1162.130	.01250	5,000
4.617	9.401	159	159.367	1003.068	1162.435	.01279	4,888
4.729	9.629	160	160.374	1002.366	1162.740	.01308	4,779
4.843	9.861	161	161.381	1001.664	1163.045	.01338	4,673
4.960	10.098	162	162.389	1000.961	1163.350	.01368	4,569
5.079	10.340	163	163.397	1000.258	1163.655	.01399	4,467
5.200	10.588	164	164.405	999.555	1163.960	.01430	4,368
5.324	10.840	165	165.413	998.852	1164.265	.01462	4,271
5.451	11.097	166	166.421	998.149	1164.570	.01495	4,177
5.580	11.359	167	167.429	997.446	1164.875	.01528	4,085
5.711	11.627	168	168.437	996.743	1165.180	.01562	3,996
5.845	11.900	169	169.445	996.040	1165.485	.01596	3,910
5.981	12.178	170	170.453	995.337	1165.790	.01631	3,826
6.120	12.461	171	171.461	994.634	1166.095	.01667	3,744
6.262	12.750	172	172.470	993.930	1166.400	.01704	3,664
6.408	13.045	173	173.479	993.226	1166.705	.01741	3,586
6.555	13.345	174	174.488	992.522	1167.010	.01779	3,510
6.704	13.651	175	175.497	991.818	1167.315	.01817	3,436
6.857	13.963	176	176.506	991.114	1167.620	.01855	3,365
7.013	14.281	177	177.515	990.410	1167.925	.01894	3,295
7.172	14.605	178	178.524	989.706	1168.230	.01934	3,226
7.335	14.935	179	179.533	989.002	1168.535	.01975	3,159
7.500	15.271	180	180.542	988.298	1168.840	.02017	3,093
7.668	15.614	181	181.551	987.594	1169.145	.02060	3,029
7.841	15.963	182	182.561	986.889	1169.450	.02104	2,966
8.016	16.318	183	183.571	986.184	1169.755	.02148	2,905
8.194	16.680	184	184.581	985.479	1170.060	.02193	2,846

TABLE OF
PROPERTIES OF WATER FROM 32° TO 212° FAHRENHEIT.—Continued.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
8.375	17.049	185	185.591	984.774	1170.365	.02238	2,789
8.558	17.425	186	186.601	984.069	1170.670	.02284	2,733
8.745	17.807	187	187.611	983.364	1170.975	.02331	2,678
8.936	18.196	188	188.621	982.659	1171.280	.02379	2,624
9.132	18.593	189	189.632	981.953	1171.585	.02428	2,571
9.330	18.997	190	190.643	981.247	1171.890	.02470	2,519
9.532	19.408	191	191.654	980.541	1172.195	.02529	2,469
9.738	19.827	192	192.665	979.835	1172.500	.02580	2,420
9.947	20.253	193	193.676	979.129	1172.805	.02632	2,372
10.160	20.687	194	194.686	978.424	1173.110	.02685	2,325
10.377	21.129	195	195.697	977.718	1173.415	.02740	2,279
10.597	21.579	196	196.708	977.012	1173.720	.02796	2,234
10.822	22.036	197	197.719	976.306	1174.025	.02853	2,190
11.051	22.502	198	198.730	975.600	1174.330	.02910	2,147
11.284	22.976	199	199.741	974.894	1174.635	.02967	2,105
11.521	23.458	200	200.753	974.187	1174.940	.03025	2,064
11.761	23.948	201	201.765	973.480	1175.245	.03083	2,024
12.006	24.446	202	202.777	972.773	1175.550	.03142	1,985
12.255	24.953	203	203.789	972.066	1175.855	.03201	1,953
12.508	25.468	204	204.801	971.359	1176.160	.03261	1,916
12.766	25.992	205	205.813	970.652	1176.465	.03323	1,880
13.028	26.525	206	206.825	969.945	1176.770	.03386	1,844
13.295	27.067	207	207.837	969.238	1177.075	.03450	1,809
13.568	27.619	208	208.849	968.531	1177.380	.03516	1,775
13.843	28.180	209	209.861	967.824	1177.685	.03584	1,741
14.122	28.751	210	210.874	967.116	1177.990	.03654	1,708
14.406	29.332	211	211.887	966.408	1178.295	.03725	1,676
14.700	29.9218	212	212.900	965.700	1178.600	.03797	1,644

EVAPORATIVE AND CALORIMETER TESTS OF COIL BOILERS

These tests were conducted by the Bureau of Steam Engineering of the United States Navy, under the direction of Commodore GEORGE W. MELVILLE, Engineer-in-Chief of the United States Navy and Chief of the Bureau, to whom the author is indebted for a copy of the official reports of these tests, which is here condensed and arranged expressly for this work, for the benefit and information of the student of steam engineering, as well as to afford him an opportunity for making a practical application of the rules heretofore laid down in this chapter.

The first and second series of tests were made with the coil boiler shown in the following engravings:

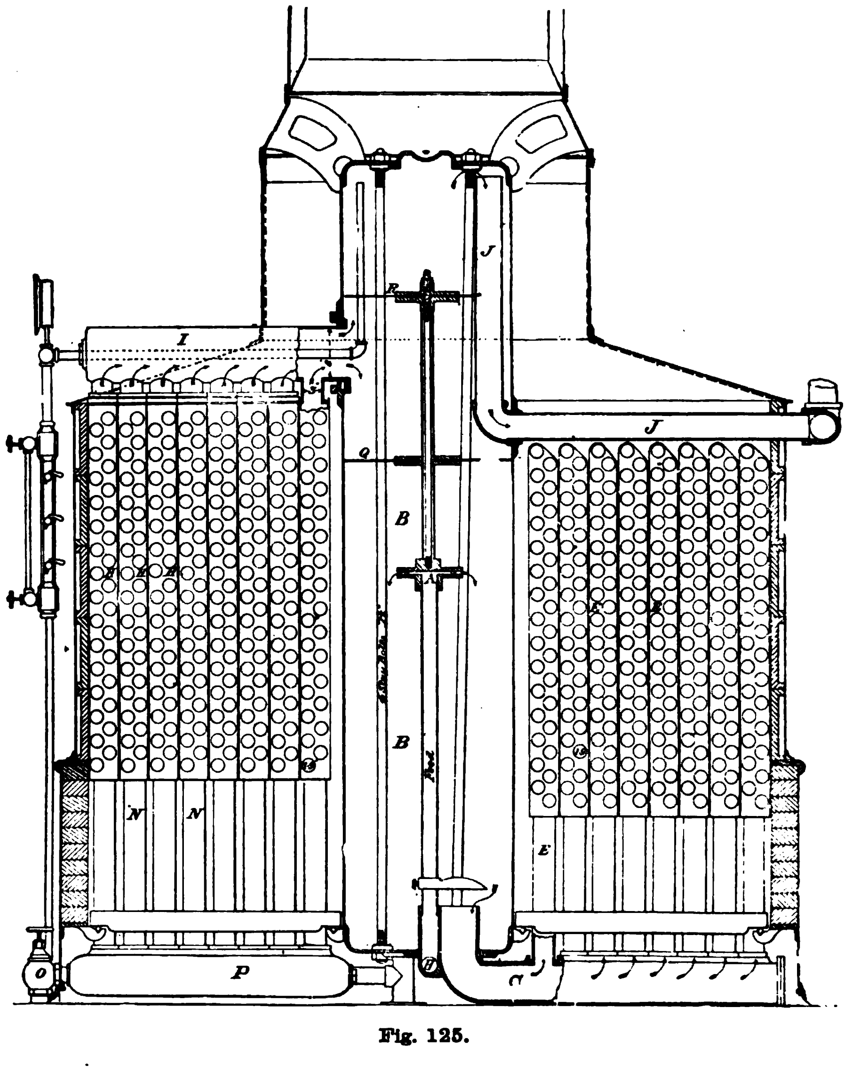


Fig. 125.

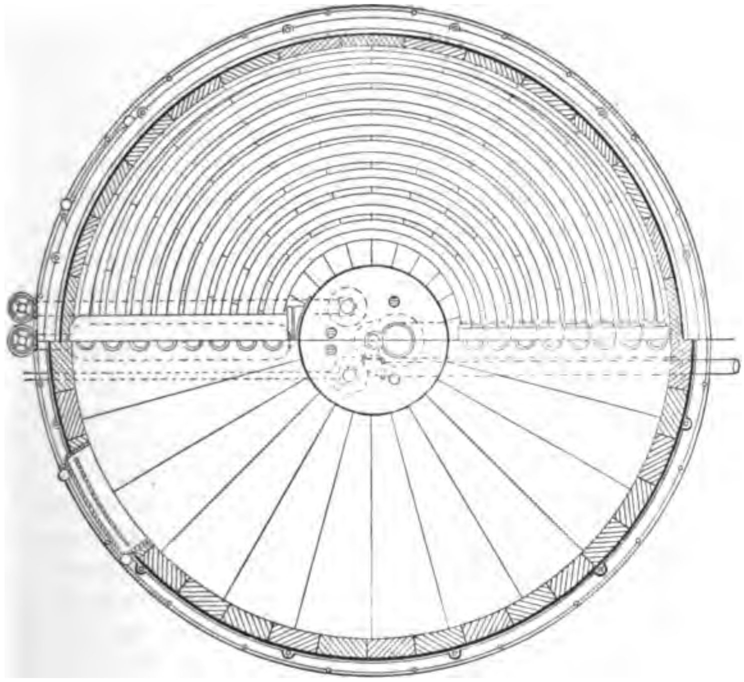


Fig. 126.

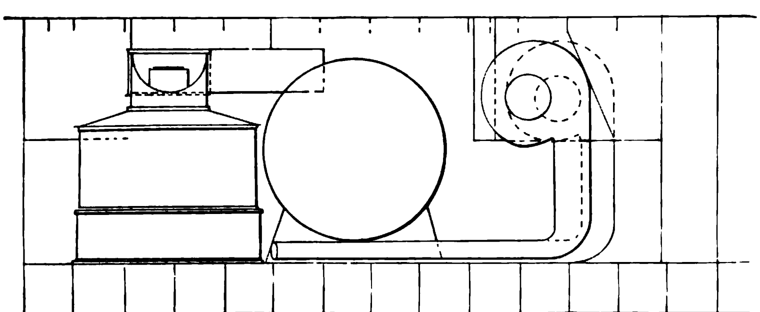


Fig. 127.

The plant with which the tests were made consisted of the boiler shown in the engravings, including two cylindrical iron feed tanks of large dimensions. One was placed somewhat above the level of the ground and through its bottom the interior was connected by a suitable pipe with the feed pump. This was called the feed tank. Above and back of this tank was placed the measuring tank. When filled to the height of the lower sides of three rectangular slits cut equidistant in its circumference near the top, it was found to contain by the scales, 4413 pounds of water at a temperature of 56° Fahrenheit. Corrected for the mean temperature of the feed water during the test (50.4°) this weight becomes 4414.4 pounds, which is the weight used in the computations. All parts of these tanks, their pipes and cocks, and the feed apparatus throughout, were in full view; all leaks were therefore visible. There were but two, and the water from these were caught and returned to the tank. The supply of water came from a large elevated tank, kept full by a double-acting plunger steam pump.

Forced draught was employed, and the air from a blower was discharged into an air-tight fire room which inclosed one-half of the circumference of the boiler for a height of 7½ feet. The air duct entered on one side of the boiler, the lower edge of its section being at the level of the floor. The air lock was shut off from the fire room by a flap door opening upwards, and from the outside atmosphere by a sliding door. Within the air lock was the scale for weighing the coal.

Two pyrometers were placed in the smoke pipe at right angles with each other and about two feet above the central drum of the boiler. The thermometer in the steam pipe stood in a bath of oil and was covered in that part projecting above the pipe.

The calorimeter was of the ordinary barrel variety, fitted with an agitator for equalizing the temperature of the water within it. Steam came to it through a pipe about 10 feet in length, which was well protected against loss of heat by radiation. It got its steam through a perforated tube inside of the steam pipe close to the boiler.

The boiler has a plate-iron central drum, standing vertically in the center of the system, and extending from beneath the grate to about 11 feet 6 inches above the base line. It receives all the feed water from an inside pipe that passes through its bottom and extends to near the water line. The space above the water line forms practically all the steam room. Surrounding this drum, in nearly concentric circles, are eight separate coils or sections, varying equally in dimensions from 36 inches of the inner one to 110 inches of the outer one. Each section has 30 tubes or complete circles, making 240 in all. The tubes of each section are connected in half circles by screwed joints to two vertical headers, diametrically opposite to each other. The tubes are 2 inches in external diameter, and are set at an angle of about ten

degrees with the horizontal, to give direction to the current of circulation within them. The headers carrying the lowest ends of the half circles have a common connection at their bottom ends with a manifold, which at its inner end is connected with the central drum. Through this manifold the headers are supplied with water, the upper ends being closed.

The headers, carrying the highest ends of the half circles, connect the manifold at their tops, through which all the steam generated passes into the upper end of the central drum. At their bottom ends they connect with a manifold, which serves as a mud drum, and as a blow-off for purging the boiler. The headers proper do not extend below the level of the circular tubes, the connections with the lower manifolds being through iron pipes 3½ inches in diameter, screwed into the bottom ends of the headers and joined to the manifolds by shallow stuffing boxes. These manifolds are beneath the grate, and they and the headers are of cast-steel. The grate is an *annulus*, and has an area of 65.5 square feet. This was reduced to 53 square feet for the test by bricking, for convenience of manipulating the fires.

The total heating surface, measured on the exterior of the tubes, headers and drum, is 2940 square feet; measured on the interior it is 2074.5 square feet. A portion of this heating surface was, however, covered by the bricks laid on the grate, so that the area of these superfices was reduced to 2473.5 and 2060 square feet, respectively. These are the figures used to represent the heating surface in making the various computations. About 30 per cent. of this heating surface is above the level of the mean water line.

The greatest diameter of the outer casing is 123 inches; the diameter of the outer casing around the drum above the tubes is 54 inches; and the diameter of smoke pipe is 40 inches. The height from base to top of drum casing is 11 feet 8 inches, and from base to top of casing around tubes is 8 feet 2 inches. The light iron casing that envelopes the furnace and tubes is lined at the furnace level with fire bricks, and above that with curved tile hollowed in the back.

Then we have:

Weight of boiler exclusive of smoke pipe, $11\frac{1898}{2240}$ tons.
Weight of boiler with water to first gauge, $13\frac{1854}{2240}$ tons.
Weight of boiler with water to second gauge, $13\frac{1906}{2240}$ tons.

The following is an analysis of the coal used in making the test:

Fixed carbon, 70.67 per cent.
Volatile matter, 25.35 per cent.
Sulphur, 0.53 per cent.
Moisture, 1.35 per cent.
Ashes, 2.10 per cent.

EVAPORATIVE TESTS.

Two tests each of twelve hours' duration were made. Beginning with cold water in the boiler, a wood fire was started and continued until steam at 160 pounds had blown off for about ten minutes. The embers were then hauled out, and with 200 pounds of wood, a new fire was started; the height of the water in the gauge glass marked, and the trial held to have commenced. The blower was run slowly until the coal was well ignited, and then at full speed. The record of the water used was made when that emptied from the measuring tank had been pumped into the boiler. At the end of the test the water remaining in the feed tank, after bringing the height in the gauge glass to the original place, was deducted from the quantity of record.

The feed water temperatures were obtained from the feed tank. The coal was filled into a barrel on the platform scales in the air lock to a fixed weight, tallied and dumped, without wheeling, into the fire room. The air pressure was maintained constant by slight variations in the speed of the blower engine. When nearing the end of a run, the fire was allowed to burn down, the blower being gradually slowed until nothing was left but clinkers giving off but little heat. When no more steam formed, the trial came to an end.

The steam generated was blown into the atmosphere through a screw top valve, the hand wheel of which was in charge of an attendant, who regulated the flow to maintain a constant pressure of 160 pounds by the gauge.

The data of the runs of record will be found in the following tables.

TABLE OF EVAPORATIVE TESTS.

Reference Number.	TIME.	Steam Pressure by Gauge at Boilers.	Barometer in Inches of Mercury.	Pounds of Coal Consumed	Pounds of Dry Refuse.	Pounds of Water Fed to Boiler.	TEMPERATURE. °FAH. Feed Water t_1	Steam by Thermometer at Boiler.	In Uptake.	Atmosphere.	Fire Room.	Air Pressure in Inches of Mercury.	REVOLUTIONS OF ENGINE AND BLOWER. Engine.	Blower.
FIRST EVAPORATIVE TEST.														
1	12:00	160	29.40	*80	.....	13,243.2	52.00	360	788	66	70	2	250	500
2	1:00	160	29.40	3,000	.....	17,657.6	51.00	360	826	67	70	2	300	600
3	2:00	160	29.40	......	.....	17,657.6	51.00	358	841	69	71	2	302	604
4	3:00	160	29.40	......	.....	17,657.6	51.00	358	823	69	74	2	294	588
5	4:00	160	29.35	6,250	.....	22,072.0	51.00	357	924	69	75	2	285	570
6	5:00	160	29.40	......	.....	22,072.0	51.00	360	912	68	74	2	285	570
7	6:00	160	29.40	7,500	.....	22,072.0	50.5	359	835	68	77	2	275	550
8	7:00	160	29.40	......	.....	17,657.6	50.25	360	821	68	73	2	280	560
9	8:00	160	29.40	8,000	.....	22,072.0	50.5	366	927	65	71	2	277	554
10	9:00	160	29.40	......	.....	17,657.6	50.00	370	897	60	70	2	275	550
11	10:00	160	29.40	......	.....	22,072.0	50.00	369	922	61	71	2	285	570
12	10:35	160	29.40	8,000	1,590	8,828.8	50.00	367	820	62	74	2	200	400
Means and Totals......			29.3958	32,830	1,590	220,720.0	50.7	362	861	66	72.5	..	275.666	551.333
SECOND EVAPORATIVE TEST.														
1	10:00	160	29.45	*80	.....	4,414.4	50.00	362	860	57	74	2	270	540
2	11:00	160	29.50	.....	.....	22,072.0	50.00	364	910	63	75	2	293	586
3	12:00	160	29.45	9,250	.....	17,657.6	50.00	366	920	69	77	2	287	574
4	1:00	160	29.45	......	.....	22,072.0	50.25	370	867	69	77	2	275	555
5	2:00	160	29.40	......	.....	22,072.0	51.00	368	972	72	79	2	262	524
6	3:00	160	29.40	9,750	.....	22,072.0	51.00	365	925	73	80	2	280	560
7	4:00	160	29.45	......	.....	22,072.0	50.50	367	930	71	81	2	260	520
8	5:00	160	29.45	......	.....	22,072.0	50.50	367	920	67	80	2	265	530
9	6:00	160	29.45	10,000	.....	17,657.6	50.00	367	908	67	79	2	272	544
10	7:00	160	29.45	......	.....	22,072.0	50.00	367	905	68	78	2	260	520
11	8:00	160	29.45	......	.....	17,657.6	49.75	367	850	65	75	2	264	528
12	9:00	160	29.45	......	.....	22,072.0	49.25	368	885	67	75	2	260	520
13	9:33	160	29.45	8,112	1,798	7,201.6	49.00	...	...	61	75	1	...	...
Means and Totals......			29.446	37,192	1,798	241,164.8	50.15	366.5	903.666	66.846	77.307	..	270.666	541.333

* WOOD. 400 pounds of wood were used at the beginning of each test, and an allowance made of $\frac{4}{10}$ of a pound of coal for each pound of wood consumed.

CALORIMETER TESTS.

Calorimeter tests were made with this boiler at regular hourly intervals during the evaporative tests—twelve during the first trial and eleven during the second, making twenty-three in all; and the quality of steam was calculated from the following formula:

$$Q=\frac{1}{l}\left(\frac{W}{w}(h'-h)-(T-h')\right)$$

Q equals quality of steam, dry saturated steam being unity.

W equals weight of water originally in calorimeter.

w equals weight of water added to W by condensed steam.

T equals total heat of water due to observed pressure.

l equals latent heat of steam at observed pressure.

h equals total heat of water of initial temperature t in calorimeter.

h' equals total heat of water of final temperature.

NOTE.—The greater the weight of condensed steam (w) the smaller will be the percentage of error in the value of Q, other things being equal.

The importance or weight of an error in any one of the observed readings, the others being correct, increases in the following order: P, W, t, t', w.

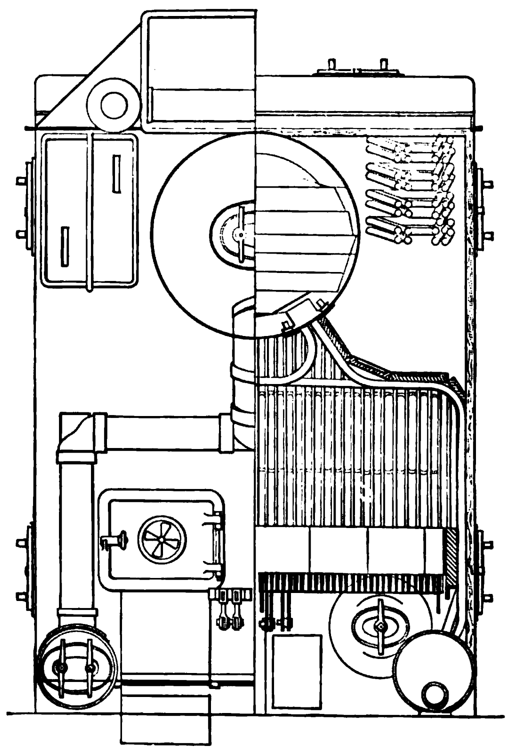


Fig. 128.

TABLE OF CALORIMETER TESTS.

Number of Test.	TIME. At Beginning.	TIME. At End.	Average Pressure of Steam in Pounds Absolute. P.	WEIGHTS. Calorimeter. b.	WEIGHTS. W+b.	WEIGHTS. W.	WEIGHTS. W+b+w.	WEIGHTS. w.	TEMPERATURE. °FAH. t′	TEMPERATURE. °FAH. t.	Q.	Temperature of Steam by Thermometer. °Fah.	
1	11:16	11:31	174.4	82	396.8	314.8	414.6	17.8	114.0	54.0	.93710	360	CALORIMETER TESTS DURING FIRST EVAPORATIVE TEST.
2	12:14	12:33	174.4	82	407.7	325.7	425.8	18.1	114.3	53.3	.98132	360	
3	1:18	1:35	174.4	82	406.9	324.9	425.7	18.8	113.0	53.0	.90730	358	
4	2:14	2:35	174.4	82	407.9	325.9	426.3	18.4	111.2	52.2	.91469	358	
5	3:20	3:37	174.4	82	404.2	322.2	422.1	17.9	110.0	52.2	.90777	357	
6	4:25	4:43	174.4	82	404.3	322.3	423.3	19.0	111.7	50.8	.88590	360	
7	5:16	5:31	174.4	82	405.0	323.0	423.6	18.6	110.6	51.2	.90765	359	
8	6:15	6:30	174.4	82	404.4	322.4	424.4	20.0	114.8	52.5	.87128	360	
9	7:20	7:39	174.4	82	404.4	322.4	423.2	18.8	110.2	51.6	.86654	365	
10	8:20	8:36	174.4	82	405.0	323.0	424.7	19.7	111.8	51.5	.84973	370	
11	9:12	9:26	174.4	82	403.4	326.4	428.2	19.8	110.2	51.8	.81730	369	
12	9:55	10:10	174.4	82	415.6	333.6	436.1	20.5	113.2	52.0	.85969	367	
* Percentage of moisture in the steam, 100 (1—Q)=11.615							Mean values of Q and temperature......				.88385	362	
1	9:55	10:14	174.4	82	413.0	331.0	433.4	20.4	112.2	50.3	.89604	362	CALORIMETER TESTS DURING SECOND EVAPORATIVE TEST.
2	10:15	11:13	174.4	82	410.0	328.0	429.3	19.3	113.8	53.7	.89029	364	
3	12:00	12:17	174.4	82	405.5	323.5	424.6	19.1	112.0	54.0	.84256	366	
4	1:00	1:20	174.4	82	407.0	325.0	426.8	19.8	113.8	53.8	.84818	370	
5	2:00	2:20	174.4	82	405.0	323.0	424.0	19.0	112.2	55.0	.83051	368	
6	3:00	3:18	174.4	82	405.0	323.0	424.6	19.6	112.8	53.2	.84258	365	
7	4:00	4:15	174.4	82	405.4	323.4	423.0	17.6	110.0	52.7	.92232	367	
8	4:57	5:14	174.4	82	408.5	326.5	427.5	19.0	111.4	53.0	.86624	367	
9	6:25	6:40	174.4	82	430.0	348.0	449.0	19.0	109.5	54.3	.87365	367	
10	7:18	7:32	174.4	82	405.0	323.0	424.6	19.6	112.0	52.2	.84627	367	
11	8:25	8:42	174.4	82	405.0	323.0	423.6	18.6	110.8	52.6	.87363	367	
* Percentage of moisture in the steam, 100 (1—Q)=13.349							Mean values of Q and temperature......				.86651	366.66	

* There is no reason why the test should show such a large percentage of moisture in the steam for this boiler, unless it be the sudden opening of the throttle valve, causing the lifting of a quantity of water from the boiler and discharging it into the calorimeter.

From the foregoing tables the following data are obtained, from which the computations for the potential evaporation are made:

Average steam pressure (absolute) P	174.4
Average temperature of feed water	50.4°
(a) Number of pounds of water vaporized, $W_1 \times Q$	404,232.0
(b) Number of pounds of water carried over with the steam, $W_1(1-Q)$	57,652.4
Total heat of steam at pressure P	1,194.88°
Total heat of water at temperature t_1	18.504°
(c) Units of heat required to vaporize one pound of water from a temperature t_1 and under a pressure P.	1,176.376
(c′) Units of heat required to raise the temperature of one pound of water from t_1 to the temperature due to the pressure P	324.196
(d) Units of heat required to vaporize one pound of water from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	965.80
Total heat required to vaporize the water $a \times c$	475,528,823.2320
Total heat required to raise the temperature of the water $b \times c'$	18,690,677.4704
(e) Total heat obtained from the fuel as measured by the steam discharged	494,219,500.7024
(f) Units of heat obtained per pound of fuel	7,058.0603
(g) Units of heat obtained per pound of combustible	7,416.9268
$\frac{f}{c}$ Potential evaporation per pound of fuel, from a temperature t_1 and under a pressure P	5.99983
$\frac{g}{c}$ Potential evaporation per pound of combustible, from a temperature t_1 and under a pressure P	6.30489
$\frac{f}{d}$ Equivalent potential evaporation per pound of fuel, from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	7.30799
$\frac{g}{d}$ Equivalent potential evaporation per pound of combustible, from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	7.67957

NOTE.—All weights are given in pounds and all temperatures in degrees Fahrenheit.

TOTAL QUANTITIES.

Duration of test, in hours, two of twelve hours each	24
Fuel consumed	70,022
Refuse from fuel, in dry ashes, dust and clinkers	3,388
Combustible consumed	66,634
Water fed to boiler, by tank measurement, W_1	461,884.6
Per cent. of the fuel in dry refuse, etc	4.8384

AVERAGE QUANTITIES.

Temperature of feed water, t_1	50.4°
Temperature of steam, by thermometer	364.33°
Temperature of uptake	882.833°
Temperature of atmosphere	72.48°
Temperature of fire room	75°
Barometer, in inches of mercury	29.4205
Pressure of steam at boiler, in pounds, per square inch above a perfect vacuum, 14.7+ pressure by gauge, in pounds, P,	174.4
Pressure of steam at engine, in pounds, per square inch above a perfect vacuum	......
Air pressure, in inches of water, in fire room	2
Air pressure, in inches of water	......
Revolutions of blowing engines, per minute	273.166
Revolutions of blower, per minute	546.333

RATES OF COMBUSTION.	Pounds of Fuel.	Pounds of Combustible.
Amount consumed per hour	2,917.5833	2,776.375
Amount consumed per hour per square foot of grate surface (53)	55.0487	52.3844
Amount consumed per hour per square foot of heating surface (exterior)	1.1795	1.1224

VAPORIZATION IN POUNDS OF WATER.	Per Pound of Fuel.	Per Pound of Combustible.
Apparent evaporation, by tank measurement, from a temperature t_1 and under a pressure P	6.5963	6.9318
Equivalent apparent evaporation, from and at 212° Fahrenheit, and under atmospheric pressure	8.0345	8.4429
Actual evaporation, into steam of quality Q, from a temperature t_1 and under a pressure P	5.7729	6.0665
Equivalent actual evaporation from and at 212° Fahrenheit, and under atmospheric pressure	7.0314	7.3890
Potential evaporation, or evaporation had all the heat obtained from fuel been utilized in converting the water in boiler into dry saturated steam from a temperature t_1 and under a pressure P	5.9986	6.3037
Equivalent potential evaporation from and at 212° Fahrenheit, and under atmospheric pressure	7.3063	7.6779

THIRD AND FOURTH SERIES OF TESTS.

The coil boiler employed in making these evaporative and calorimetric tests is shown in Figs. 128 and 129.

This boiler is 11 feet 5 inches long, 7 feet 9 inches in breadth, and 12 feet 1¾ inches in extreme height. It has 2026.75 square feet of heating surface, of which 9.43 per cent. is superheating surface. The grate area is 47 square feet, and the ratio of grate to heating surface is 1 to 43.12. The boiler was supplied with rocking grates, but they were

little used during the tests. The estimated weight of the boiler empty is 9.75 tons, and the actual weight of water to the second gauge is 1.8 tons. Forced draught was employed, the air ducts entering the ash-pit on one side. During the tests these air ducts were shut off when the fire doors were opened.

The plant used for the tests contained two large tanks, the measuring tank holding, when filled to the lower edge of the wier, 8990.15 pounds of water, corrected for the temperature of the feed. Their contents at different levels were ascertained by weighing into them successive 400 pound quantities, the level after each weighing being marked on a batten.

The calorimeter was of the ordinary barrel variety. Its steam came through a 1 inch pipe about 8 feet long, well covered. Steam entered this pipe through a perforated tube which entered the main pipe near the boiler.

A wood fire was started and continued until steam at 160 pounds had blown off for some minutes. The embers were then hauled out and a new fire started, the height of the water in the glass gauge marked, and the trial held to commence. The wood was entered in the table at four-tenths its weight. The speed of the blower was gradually increased until 2 inches of water column was attained. The record of the water was made when the measuring tank was emptied. At the end of the test the water remaining in the feed tank, after bringing that in the boiler to the original height in the glass, was deducted from the quantity last recorded. The feed water temperatures were taken from the feed tank. The coal was filled into a barrow on a platform scale to a fixed weight (300 pounds net) and tallied. When nearing the end of the run the firing was discontinued, the blower gradually slowed until the fire had so far burned out that no steam was formed, and the trial then came to an end. The steam generated was blown into the atmosphere through a screw stop-valve, which was controlled by an attendant, who regulated the flow to maintain a constant pressure of 160 pounds by gauge.

The calorimeter tests were made with regularity; the water in the barrel was well agitated and the top of the latter was covered. The hose was heated before being placed in the barrel, and accuracy in weight obtained by the use of counter-weights of one tenth of a pound each. The data of the runs of record will be found in the following tables:

TABLE OF EVAPORATIVE TESTS.

Reference Number.	TIME.	Steam Pressure by Gauge at Boilers.	Barometer in Inches of Mercury.	Pounds of Fuel Consumed	Pounds of Dry Refuse.	Pounds of Water Fed to Boilers.	TEMPERATURE.—°FAH. Feed Water t_1.	TEMPERATURE.—°FAH. Steam by Thermometer at Boiler.	TEMPERATURE.—°FAH. In Uptake.	TEMPERATURE.—°FAH. Atmosphere.	Air Pressure in Inches of Water.	Revolutions of Blowing Engine.
FIRST EVAPORATIVE TEST.												
1	8:19	160	30.15	{* 160 / 4,200}	.....	8,990.15	58	...	.....	..	2	380
2	9:19	160	30.15	2,400	.....	8,990.15	58	514	995	60	2	...
3	10:19	160	30.15	1,200	.....	8,990.15	58	515	1,070	62	2	...
4	11:19	160	30.15	2,100	.....	17,980.30	58	500	915	63	2	...
5	12:19	160	30.15	1,500	.....	8,990.15	58	506	1,055	64	2	...
6	1:19	160	30.15	1,500	.....	8,990.15	58	510	1,040	65	2	...
7	2:19	160	30.15	1,800	.....	17,980.30	58	515	1,035	67	2	...
8	3:19	160	30.15	2,100	.....	8,990.15	58	520	1,065	67	2	...
9	4:19	160	30.15	1,800	.....	17,980.30	58	500	1,025	69	2	...
10	5:19	160	30.15	2,100	.....	8,990.15	58	475	1,035	71	2	...
11	6:19	160	30.15	1,200	.....	17,980.30	58	480	1,005	69	2	...
12	7:50	160	30.15	......	3,457	1,840.75	58	...	825	64	2	...
Means and Totals..........				20,060	.....	136,693.00	..	503.5	1,005.5	65.54	..	...
SECOND EVAPORATIVE TEST.												
1	8:07	160	30.05	{* 160 / 3,600}	.....	8,990.15	58	...	880	60	2	380
2	9:07	160	30.05	2,100	.....	3,990.15	58	430	875	66	2	...
3	10:07	160	30.05	1,500	.....	8,990.15	58	...	865	67	2	...
4	11:07	160	30.05	2,100	.....	17,980.30	58	390	925	68	2	...
5	12:07	160	30.05	1,800	.....	8,990.15	58	380	925	68	2	...
6	1:07	160	30.05	1,800	.....	17,980.30	58	385	925	68	2	...
7	2:07	160	30.05	2,400	.....	8,990.15	58	420	925	70	2	...
8	3:07	160	30.05	2,100	.....	8,990.15	58	434	1,025	70	2	...
9	4:07	160	30.05	2,100	.....	17,980.30	58	445	925	70	2	...
10	5:07	160	30.05	1,800	.....	8,990.15	58	390	940	71	2	...
11	6:07	160	30.05	2,100	.....	17,980.30	58	394	925	69	2	...
12	6:45	160	30.05	......	2,870	{8,990.15 / 360.00}	58	...	.....	...	..	...
Means and Totals..........				23,560	.....	114,202.40	..	407.55	926	68	..	...

* Wood.

During the trial seventeen calorimetric tests were made, and the quality of steam was calculated by the following formula:

$$Q=\frac{1}{l}\left(\frac{W}{w}(h'-h)-(T-h'\right)$$

Q equals quality of steam, dry saturated steam being unity.
W equals weight of water originally in calorimeter.
w equals weight of water added to W by condensed steam.
T equals total heat of water due to observed pressure of steam.
l equals latent heat of steam at observed pressure.
h equals total heat of water of initial temperature t in calorimeter.
h' equals total heat of water of final temperature t' in calorimeter.

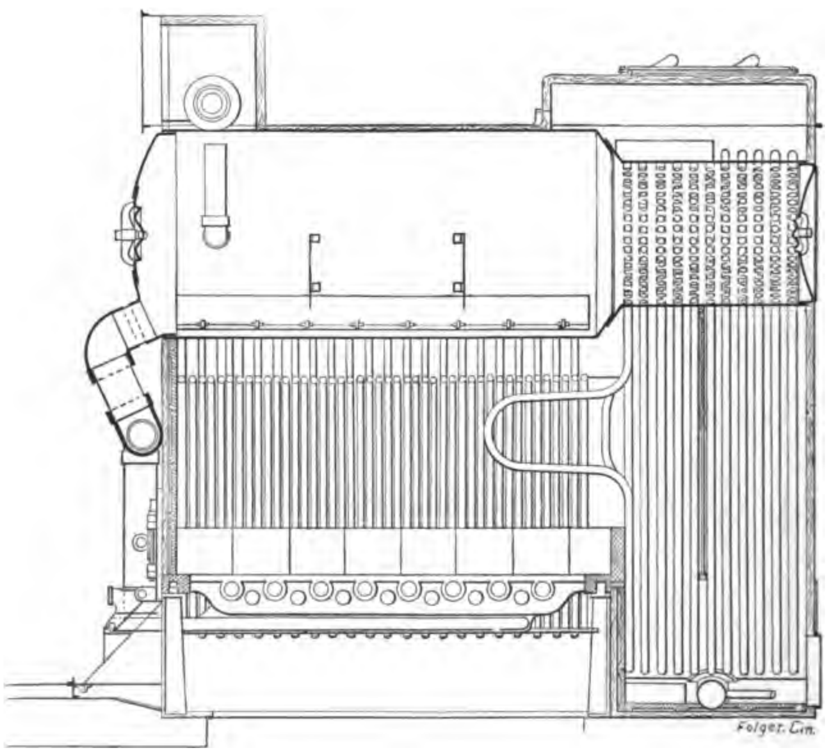


Fig. 129.

The observed data and computed results are given in the following table:

TABLE OF CALORIMETER TESTS.

Number of Test.	TIME. At Beginning	TIME. At End.	Average Pressure of Steam in Pounds Absolute. P.	WEIGHTS. Calorimeter. b.	WEIGHTS. W+b.	WEIGHTS. W.	WEIGHTS. W+b+w.	WEIGHTS. w.	TEMPERATURE. °FAH. t'	TEMPERATURE. °FAH. t.	Q.	Temperature of Steam by Thermometer. °Fah.
FIRST TEST OF SECOND TRIAL												
1	9:00	9:25	174.81	72	407.2	335.2	427.7	20.5	128.0	56.0	1.0917	514
2	9:30	9:50	174.81	72	404.2	332.2	420.7	16.5	115.6	56.9	1.0797	515
3	10:30	10:45	174.81	72	405.0	333.0	420.2	15.2	112.7	56.2	1.1420	500
4	11:27	11:45	174.81	72	404.5	332.5	420.2	15.7	111.1	56.5	1.0446	506
5	1:15	1:30	174.81	72	406.5	334.5	422.1	15.6	111.0	56.5	1.0839	515
6	3:00	3:15	174.81	72	406.2	334.0	422.2	16.0	112.0	56.5	1.0483	520
7	4:25	4:40	174 81	72	406.0	334.0	421.7	15.7	110.8	56.1	1.0529	475
8	5:30	5:48	174.81	72	404.4	332.0	420.2	15.8	111.9	56.4	1.0572	480
Percentage of moisture in the steam, 100 (1—Q)= —7.5							Mean values of Q and temperature......				1.07504	503.125
SECOND TEST OF SECOND TRIAL												
1	9:23	9:36	174.81	71.2	400.2	329.0	414.7	14.5	112.7	60.0	1.09308	430
2	10:20	10:32	174.81	71.2	400.3	329.1	415.2	14.9	113.1	60.9	1.04277	390
3	11:20	11:35	174.81	71.2	401.0	329.8	416.4	15.4	112.5	60.4	.99850	380
4	12:15	12:25	174.81	71.2	401.8	330.6	416.9	15.1	112.5	60.4	1.02770	385
5	1:30	1:40	174.81	71.2	401.2	330.0	417.1	15.9	115.1	60.6	1.05980	420
6	2:40	2:50	174.81	71.2	400.4	329.2	415.1	14.7	113.0	61.0	1.04445	434
7	3:40	3:50	174.81	71.2	400.0	328.8	414.9	14.9	113.1	61.0	1.03880	445
8	4:40	4:50	174.81	71.2	400.1	328.9	415.5	15.4	113.6	60.8	1.01377	390
9	5:40	5:49	174.81	71.2	400.0	328.8	416.1	16.1	116.2	60.7	1.02370	394
Percentage of moisture in the steam, 100 (1—Q)= —3.8							Mean values of Q and temperature......				1.038063	407.55

From the foregoing tables the following data are obtained, from which the computations for the potential evaporation are made:

Average steam pressure (absolute) P	174.81
Average temperature of the feed water, t_1	58°
(*a*) Number of pounds of water vaporized, $W_1 \times Q$	280,822.4
(*b*) Number of pounds of water carried over with the steam $W_1 (1-Q)$	0°
Total heat of steam at pressure P	1,194.962
Total heat of water at temperature t_1	26.12
(*c*) Units of heat required to vaporize one pound of water from a temperature t_1 and under a pressure P	1,168.962
(c_1) Units of heat required to raise the temperature of one pound of water from t_1 to the temperature due to the pressure P	316.785
(*d*) Units of heat required to vaporize one pound of water from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	965.8
Total heat required to vaporize the water $a \times c$	328,275,000
Total heat required to raise the steam from the temperature of saturation to actual temperature	18,209,252.1
(*e*) Total heat obtained from the fuel, as measured by the steam discharged	346,484,252.1
(*f*) Units of heat obtained per pound of fuel	7,595
(*g*) Units of heat obtained per pound of combustible	8,817.96
$\frac{f}{c}$ Potential evaporation per pound of fuel from a temperature t_1 and under a pressure P	6.4972
$\frac{g}{c}$ Potential evaporation per pound of combustible from a temperature t_1 and under a pressure P	7.5434
$\frac{f}{d}$ Equivalent potential evaporation per pound of fuel from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	7.8639
$\frac{g}{d}$ Equivalent potential evaporation per pound of combustible from and at a temperature of 212° Fahrenheit, and under atmospheric pressure	9.1302

NOTE.—All weights are given in pounds and all temperatures in degrees Fahrenheit.

TOTAL QUANTITIES.

Duration of test in hours	24.15
Fuel consumed	45,620
Refuse from fuel in dry ashes, dust and clinkers	6,327
Combustible consumed	39,293
Water fed to boiler, by tank measurement, W_1	280,822.4
Per cent. of the fuel in dry refuse, etc	13.87

AVERAGE QUANTITIES.

Temperature of feed water t_1	58°
Temperature of steam, by thermometer	458.053°
Temperature of uptake	965.909°
Temperature of atmosphere	66.682°
Temperature of fire room	
Barometer, in inches of mercury	30.10
Pressure of steam at boiler in pounds, per square inch above a perfect vacuum, 14.7+ pressure by gauge in pounds, P	174.7
Pressure of steam at engine, in pounds, per square inch above a perfect vacuum	......
Air pressure, in inches of water	2
Revolutions of blowing engines, per minute	382

RATES OF COMBUSTION.	Pounds of Fuel.	Pounds of Combustible.
Amount consumed per hour	1,889.026	1,627.039
Amount consumed per hour per square foot of grate surface	40.192	34.617
Amount consumed per hour per square foot of heating surface (exterior)	.93204	.80278

VAPORIZATION IN POUNDS OF WATER.	Per Pound of Fuel.	Per Pound of Combustible.
Apparent evaporation, by tank measurement, from a temperature t_1 and under a pressure P	6.156	7.147
Equivalent apparent evaporation, from and at 212° Fahrenheit, and under atmospheric pressure	7.450	8.650
Actual evaporation into steam of quality Q, from a temperature t_1 and under a pressure P	6.156	7.147
Equivalent actual evaporation, from and at 212° Fahrenheit, and under atmospheric pressure	7.450	8.650
Potential evaporation, or evaporation had all the heat obtained from fuel been utilized in converting the water in the boiler into dry saturated steam from a temperature t_1 and under a pressure P	6.49747	7.54344
Equivalent potential evaporation from and at 212° Fahrenheit, and under atmospheric pressure	7.86325	9.12915

NOTE.—The heat units in the foregoing computations are reckoned from 32° Fahrenheit. Peabody's tables were employed in making these tests.

EVAPORATIVE AND CALORIMETER TESTS OF LAND BOILERS.

A practical test of two land boilers, in which the author participated, is here illustrated for the information of the student of steam engineering. This test was not made to determine the efficiency of the boilers, but to determine the economy and efficiency of the Hawley Down Draft Furnace, which was attached to the boilers. Three things were to be determined: The efficiency of the furnace in increasing the horse power of the boilers; the efficiency of the furnace in increasing the economy of the boilers; and the efficiency of the furnace in preventing the formation of smoke; in all of which the furnace showed

a most remarkable performance, as shown by the following data of the test made.

The trial covered a period of ten hours and five minutes, beginning at nine o'clock, A. M., and ending at seven o'clock and five minutes, P. M., during which the following observations were carefully made:

Pressure of atmosphere by barometer.

Boiler pressure by gauge.

Force of chimney draft by gauge connected with breeching at the base of the chimney.

Temperature of waste gases by pyrometer inserted in breeching over the boilers.

Temperature of atmosphere, in open air and in boiler room, by thermometers suitably located.

Temperature of feed water by thermometers, with their bulbs suspended in brass wells containing oil; each thermometer being inserted in a feed pipe close to the boiler.

Quality of steam was determined by a Barrus Universal Calorimeter, closely connected with and piped from a nozzle located over the center of the steam drum of the boilers.

All instruments, except the barometer, were observed and readings taken every fifteen minutes during the trial.

The barometer was read hourly.

The calorimetric condition of the steam was taken every five minutes.

The scales employed for weighing coal, and scales employed for weighing feed water, were carefully tested before commencement of the trial and after its completion.

A steam gauge was employed that accurately registered the steam pressure.

The accumulation of water from the drip box of the calorimeter was accurately weighed from time to time.

The steam discharged per hour, by the orifice in the heat gauge of the calorimeter, was computed by the following formula:

$$\left(\frac{A \times P \times S}{70}\right) \times C = X. \quad \text{Pounds of steam discharged per hour.}$$

A equals area of orifice in heat gauge of calorimeter, and in this case equaled .012271875 square inch.

P equals absolute pressure due the temperature of the steam shown by the upper thermometer of the calorimeter.

S equals number of seconds in one hour.

70 equals a constant.

C equals .9682, a constant.

X equals number of pounds of steam discharged per hour.

Putting the formula in words we have the following rule:

RULE.—Multiply the area of the orifice in the heat gauge of the calorimeter by the absolute steam pressure per square inch (gauge and atmospheric pressure added together) due the temperature of the steam shown by the upper thermometer of the calorimeter, then multiply the product by 3600, *the number of seconds in one hour, then divide the last product by the constant* 70, *and finally, multiply the quotient by the constant* .9682, *and the product of this operation will give the number of pounds of steam discharged per hour by the orifice of the heat gauge of the calorimeter.*

Example.—Let 125 one thousandths of an inch ($\frac{1}{8}$ inch) equal diameter of orifice of the heat gauge.

Let .7854 equal a constant.

Let 83.76 pounds equal absolute pressure per square inch due the average temperature of steam shown by the thermometer of the heat gauge.

Let 3600 equal number of seconds in one hour.

Let 70 equal a constant.

Let .9682 equal a constant.

Then we have:

$$\left(\frac{.125^2\times.7854\times83.76\times3600}{70}\right)\times.9682=51.1819566 \text{ lbs.}$$ Steam discharged per hour by orifice of heat gauge of calorimeter.

Performing the operation, we have:

```
         .125         Diameter of orifice.
         .125         Diameter of orifice.
         ----
          625
         250
        125
      -------
      .015625         Square of diameter of orifice.
        .7854         A constant.
      -------
        62500
       78125
      125000
     109375
  -----------
 .0122718750          Area of orifice.
        83.76         Steam pressure per square inch.
  -----------
   736312500
  859031250
 368156250
981750000
-------------
1.027892250000
             3600     Number of seconds in one hour.
----------------
 616 735350000000
3083 676750000
----------------
3700.412100000000     "The Product."
```

Next, dropping the ciphers in the decimals and dividing the product by 70, we have:

```
70) 3700.4121 (52.863+  "The Quotient."
    350
    ---
     200
     140
     ---
      604
      560
      ---
       441
       420
       ---
        212
        210
        ---
         21
```

Finally, multiplying the quotient by the constant .9682, we have:

```
     52.863
      .9682
   --------
     105726
    422904
   3 17178
  47 5767
 ----------
 51.1819566  Pounds of steam discharged per hour by
             orifice of heat gauge of calorimeter.
```

Whence the percentage of moisture in steam intercepted by the drip box of the calorimeter becomes:

$$\frac{W}{X} \times 100 = M, \text{ in which}$$

W equals weight of water drawn from the drip box per hour.

X equals weight of steam discharged by orifice of heat gauge per hour.

M equals percentage of moisture of steam.

Putting the formula in words, we have the following rule:

RULE.—Divide the weight of water in pounds, intercepted by the drip box per hour, by the total weight of steam in pounds discharged by the orifice of the heat gauge per hour, and multiply the quotient by 100, *and the product will give the percentage of the moisture in the steam.*

Example.—Let 8 tenths of a pound equal weight of water intercepted by drip box per hour.

Let 51.1819 pounds equal weight of steam discharged by orifice of heat gauge per hour.

Let 100 equal a constant.

Then we have:

$$\left(\frac{.8}{51.1819}\right)\times 100=1.563 \text{ Percentage of moisture in steam received by the drip box}$$

Performing the operation, we have:

```
51.1819) .800000 (0.01563+   "The Quotient."
          511819
          -------
          2881810
          2559095
          --------
           3227150
           3070914
           --------
            1562360
            1535457
            -------
```

Multiplying "The Quotient" by 100, we have:

```
 .01563
    100
 -------
1.56300   Percentage of moisture in steam received by the drip box.
```

The formula for computing the moisture remaining in the steam after leaving the drip box, and indicated by the thermometer of the heat gauge, is given by Barrus, the inventor of the calorimeter employed in this trial, to be:

$$\frac{T-\overline{{}^{\circ}t^{\circ}}}{C}=\text{percentage of moisture, when}$$

T equal normal reading of lower thermometer.

$\overline{{}^{\circ}t^{\circ}}$ equal reading of lower thermometer while operating boilers.

C equal 21.2 a co-efficient for temperature between 310° and 320° Fahrenheit, given by upper thermometer.

Putting the formula in words, we have the following rule:

RULE.—Subtract the reading of the lower thermometer while operating the boilers from the normal reading of the lower thermometer, and divide the remainder by the constant, which in this case is 21.2, *and the quotient will give the percentage of moisture in steam after leaving drip box.*

Example.—Let 266° equal normal temperature by lower thermometer.

Let 265.6 equal average temperature by lower thermometer.

Let 21.2 equal a co-efficient for temperatures between 310° and 320°.

Then we have: $\frac{266-265.6}{21.2}=.018+$ of one per cent.

Performing the operation, we have:

266	Normal temperature by lower thermometer.
265.6	Average temperature by lower thermometer.
.4	"The Remainder."

Next, dividing The Remainder by the co-efficient 21.2, we have:

```
21.2) .400 (0.018+ of one per cent.
      212
      ----
      1880
      1696
      ----
```

MOISTURE IN COAL—HOW DETERMINED.

The moisture in the coal was determined by carefully weighing 100 pounds of coal in a bag at the commencement of the trial, and placing it on the shell of one of the boilers to remain during the trial to be thoroughly dried. At the end of the trial it was again weighed and reduction in weight determined and carefully noted.

The smoke stack was frequently noticed during the trial, especially during the firing of the furnaces and the breaking up of the fires, and during the several tests scarcely any smoke was emitted from the stack, and at no time was there any black or dense smoke whatever, notwithstanding the crowding of the boilers to their utmost capacity.

KIND OF BOILERS—HORIZONTAL TUBULAR.

Number of boilers, 2.

Diameter of boilers, 72 inches.

Length of boilers, 18 feet.

Number of tubes in each boiler, 80.

Diameter of tubes, 4 inches.

Length of tubes, 18 feet.

Length of grate surface, 4 feet.

Width of grate surface (both boilers), 13.83 feet.

Area of grate surface, 55.32 square feet.

Air space in grate surface, 52.80 per cent.

Diameter of stack (sheet iron), 48 inches.

Height of stack above grates, 88 feet.

Effective heating surface—two-thirds of shell and two-thirds of tube surface being computed as effective—2460 square feet.

Ratio of effective heating surface to one foot of grate surface 44.76 square feet.

Nominal horse power of both boilers, 204 horse power.

CALORIMETER EXPERIMENTS.

Duration of calorimeter experiments, 8.25 hours.
Average temperature by upper thermometer, 315° Fahrenheit.
Average temperature by lower thermometer, 265° Fahrenheit.
Normal temperature by lower thermometer, 266° Fahrenheit.
Total moisture intercepted by drip box, 6.62 pounds.

$$\frac{6.62}{8.25}=.8+\text{ lbs.}$$ Average moisture intercepted by drip box per hour.

$$\left(\frac{.02789225\times 83.76\times 3600}{70}\right)\times .9682=51.18+\text{ lbs.}$$ Average weight of steam escaped through orifice of heat gauge per hour.

$$\left(\frac{.8}{51.18}\right)\times 100=1.56+\text{ per cent.}$$ Moisture received by drip cup per hour.

$$\frac{266-265.6}{21.2}=.018+\text{ of 1 per cent.}$$ Moisture in steam after leaving drip box and entering heat gauge.

$$1.56+.018=1.578\text{ per cent.}$$ Total entrainment of water in steam.

OBSERVED DATA AND COMPUTED RESULTS.

Duration of trial, 10.083 hours.
Average pressure by gauge, 87.70 pounds.
Average atmospheric pressure by barometer, 14.49 pounds.
Average absolute steam pressure, 103.19 pounds.
Average temperature of feed water, 90.27° Fahrenheit.
Total weight of coal fired on grate, 16925 pounds.
Moisture in coal, 1 per cent.
Total weight of dry coal fired, 16,755.75 pounds.
Total weight of clinkers and ashes, 1351 pounds.
Total weight of combustible consumed, 15,404.75 pounds.
Average condition of steam (dry steam equals 1) .9842.
Total weight of water evaporated, 140,033 pounds.

CAPACITY RESULTS.

$$140033\times .9842=137820.5\text{ lbs.}$$ Water actually evaporated corrected for quality of the steam

$$\left(\frac{1214.5-90.27}{965.7}\right)\times 137820.5=160436.8+\text{ lbs.}$$ Equivalent evaporation as from and at 212° Fah.

$$\frac{160436.8}{10.083}=15911.6+\text{ lbs.}$$ Equivalent evaporation per hour from and at 212° Fahrenheit.

ECONOMIC EVAPORATION.

$$\frac{137820.5}{16755.75}=8.225+\text{ lbs.}$$ Water actually evaporated per pound of dry coal from actual steam pressure and temperature of feed water.

$$\frac{160436.8}{16755.75}=9.575+\text{ lbs.}$$ Equivalent evaporation of water per pound of dry coal from and at 212° Fahrenheit.

$$\frac{160436.8}{15404.7}=10.41+\text{ lbs.}$$ Equivalent evaporation of water per pound of combustible from and at 212° Fahrenheit.

RATE OF EVAPORATION

$$\frac{15911.6}{2460}=6.46+\text{ lbs.}$$ Water evaporated from and at 212° Fahrenheit per square foot of effective heating surface per hour.

COMMERCIAL HORSE POWER.

The Centennial standard of horse power for a boiler is the evaporation of 30 pounds of water per hour per horse power, at a steam pressure of 70 pounds per square inch, and feed water at a temperature of 100° Fahrenheit. To reduce this to equivalent evaporation as from and at 212° Fahrenheit, the standard adopted by the American Society of Mechanical Engineers, proceed according to the following rule:

RULE.—From the total heat units in steam at 70 *pounds pressure, subtract the heat units in the feed water at* 100° *Fahrenheit, then divide the remainder by* 965.7, *and multiply the quotient by* 30 (*the Centennial standard*), *and the product will give the commercial standard evaporation from and at* 212° *Fahrenheit for one horse power.*

Example.—Let 1210.2423 equal heat units in steam at 70 pounds pressure.
Let 100.08 equal heat units in feed water at 100° Fahrenheit.
Let 965.7 equal a constant.

Then we have:

$$\left(\frac{1210.2423-100.08}{965.7}\right)\times 30=34.48770+\text{lbs.}$$ Water evaporated per commercial horse power per hour, from and at 212° Fahrenheit.

Performing the operation, we have:

$$\begin{array}{r} 1210.2423 \\ 100.08 \\ \hline 1110.1623 \end{array}$$

Am't carried forward, 1110.1623

Am't brought forw'd, 965.7) 1110.1623 (1.14959+
965 7 | 30
144 46 | 34.48770 lbs. Water evaporated per commercial horse power per hour, from and at 212° Fahrenheit.
96 57
47 892
38 628
9 2643
8 6913
57300
48285
90150
86913

The Centennial standard, when reduced to equivalent evaporation as from and at 212° Fahrenheit, is very nearly 34.5 pounds of water evaporated per horse power per hour, which is the standard commercial horse power.

Applying that standard to the case in point, we have:

$$\frac{15911.6}{34.5}=461.2+$$ Horse power developed by the boilers in the case under consideration.

Nominal horse power of the boilers, on the basis of 12 square feet of effective heating surface per horse power:

$$\frac{2460}{12}=205$$ Nominal horse power of the two boilers.

Commercial horse power developed over nominal horse power of the boilers:

$$\left(\frac{461.2-205}{205}\right)\times 100=125-\text{ per cent.}$$

Comparisons were made with an evaporative and calorimeter test, made with a battery of two tubular boilers, the exact duplicate of these, except that those with which comparisons were made were equipped with the ordinary furnace, and in point of economy, those equipped with the Hawley furnace showed a reduction of 35.6 per cent. in the cost of fuel, and an increase in commercial horse power developed, as compared with the other boilers, of 66.5 per cent., and showed an entire abatement of the smoke nuisance; all of which was accomplished with the use of a very inferior quality of bituminous slack and nut coal.

In order that the student may get a clear idea of the construction of this furnace, the following engravings are provided for his careful study:

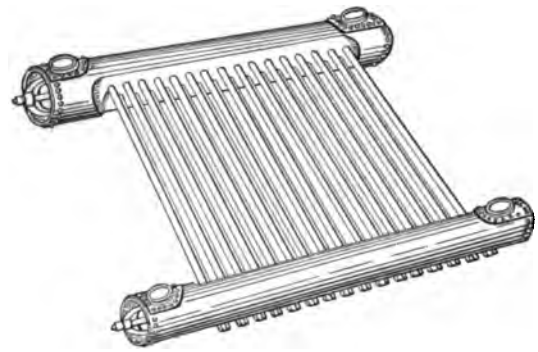


Fig. 130.

HAWLEY DOWN DRAFT WATER TUBE GRATE.

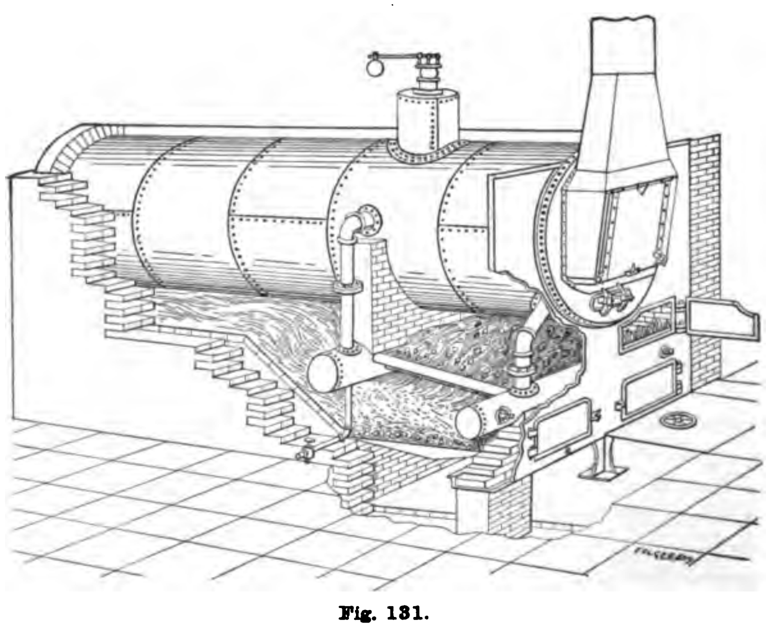


Fig. 131.

TUBULAR BOILER.

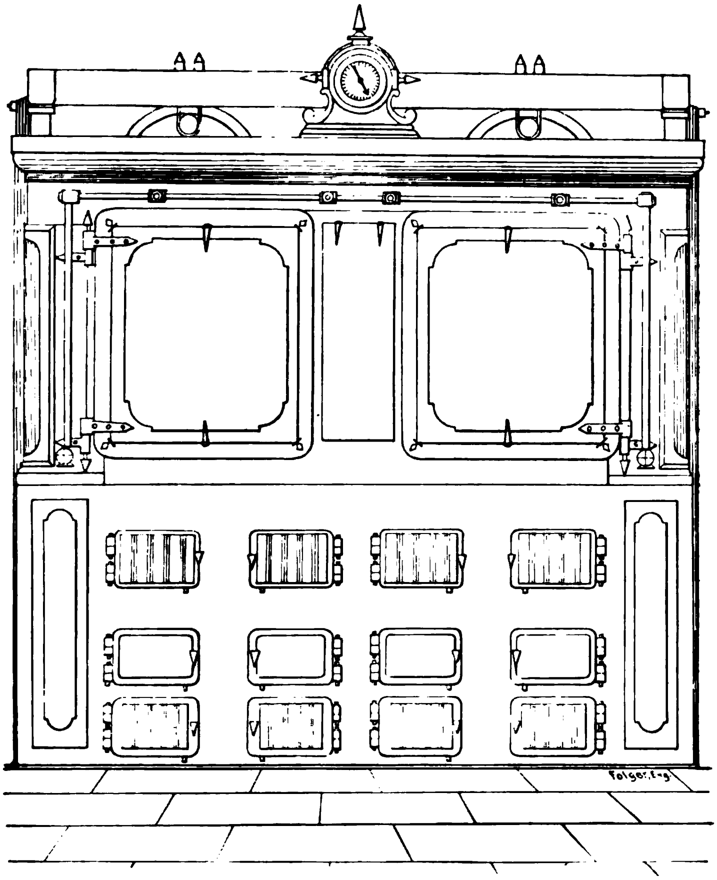


Fig. 132.
HAWLEY DOWN DRAFT FURNACE FRONT.

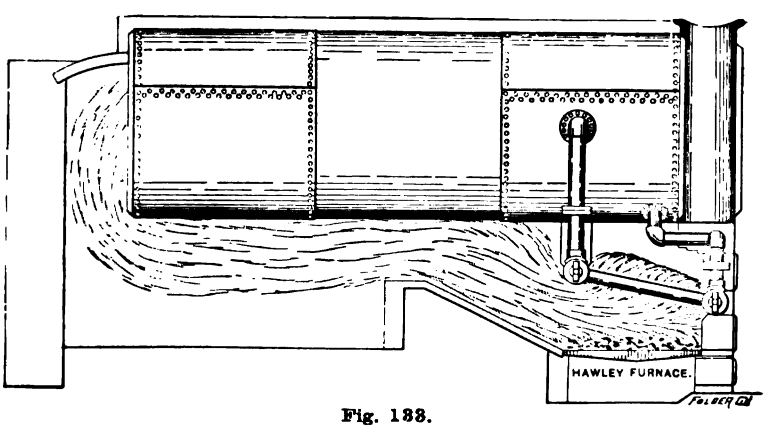


Fig. 133.
TUBULAR BOILER.

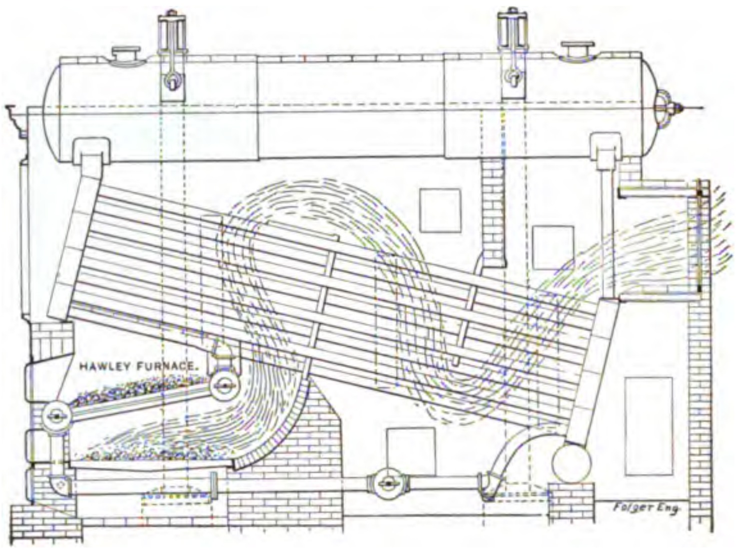


Fig. 134.

NATIONAL WATER TUBE BOILER.

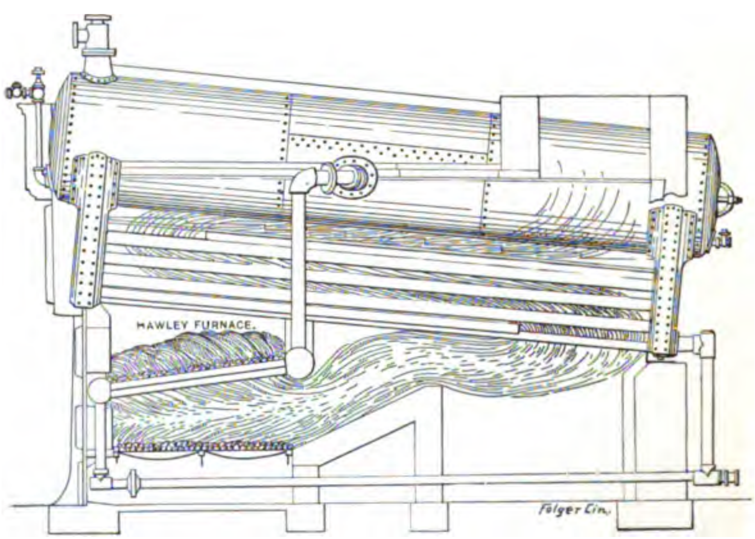


Fig. 135.

HEINE BOILER.

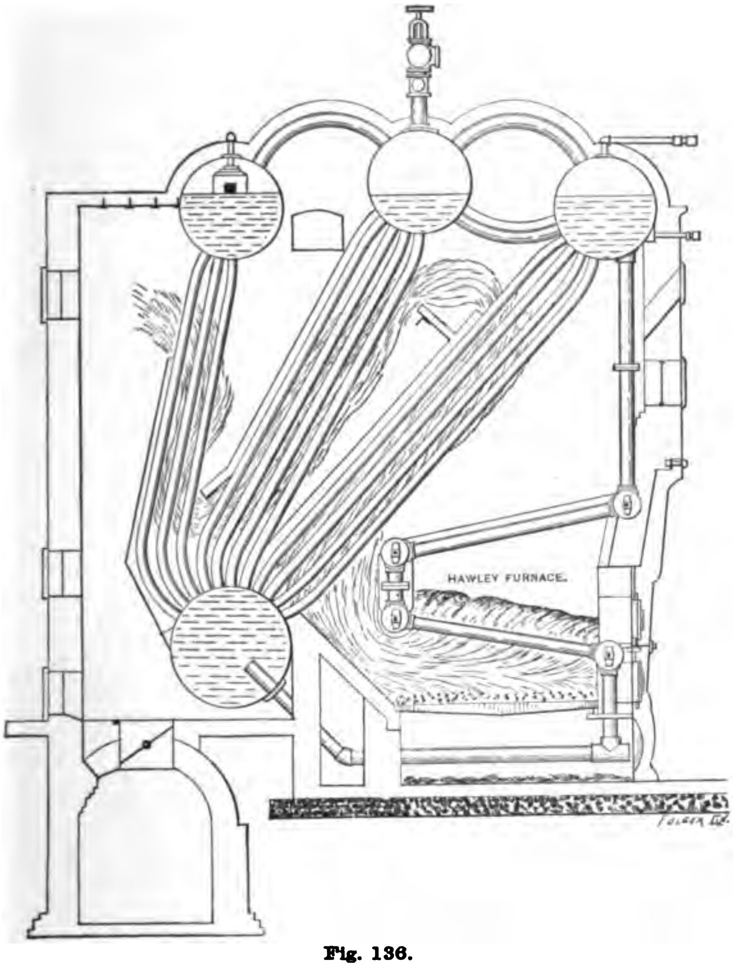


Fig. 136.
STIRLING BOILER.

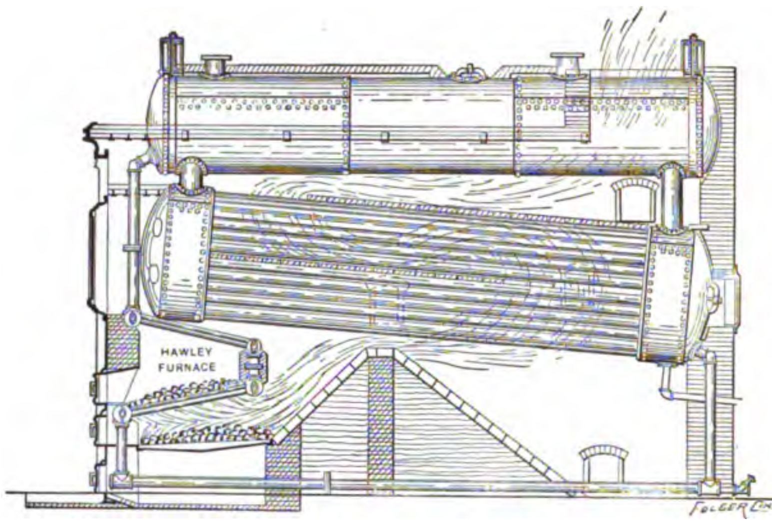


Fig. 137.

MUNROE WATER TUBE BOILER.

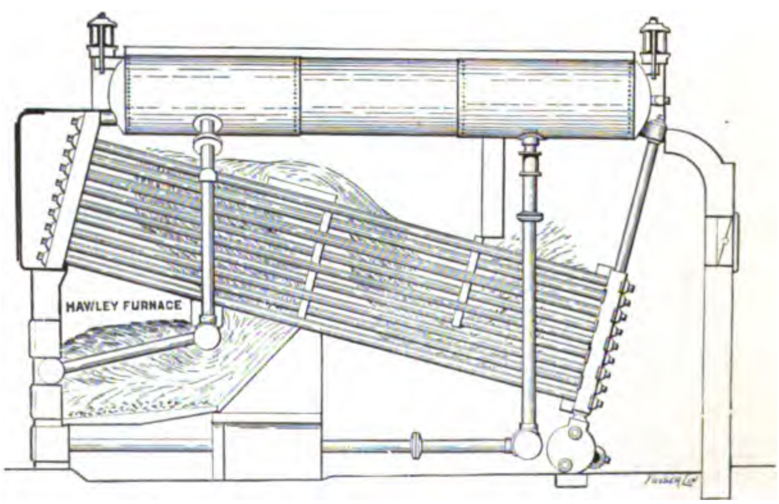


Fig. 138.

BABCOCK & WILCOX BOILER

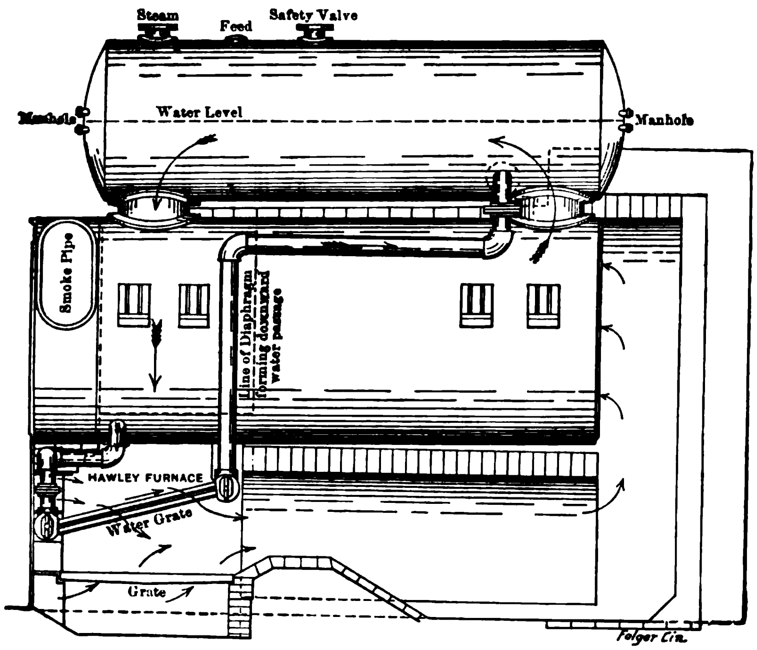


Fig. 139.

PECK BOILER.

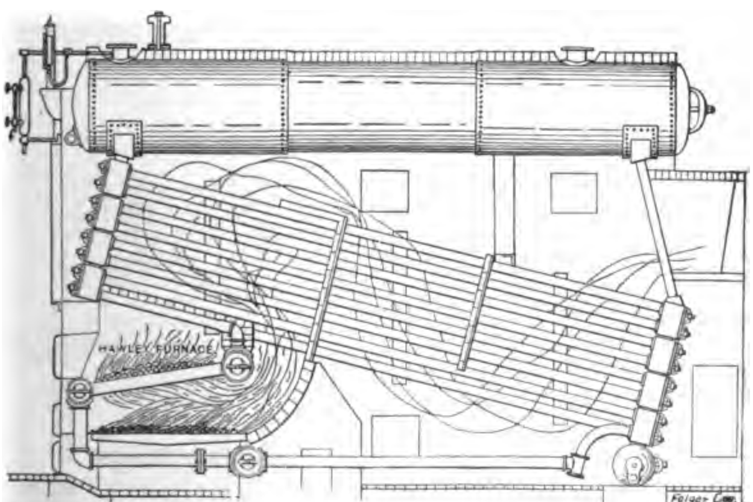


Fig. 140.

CALDWELL STANDARD BOILER.

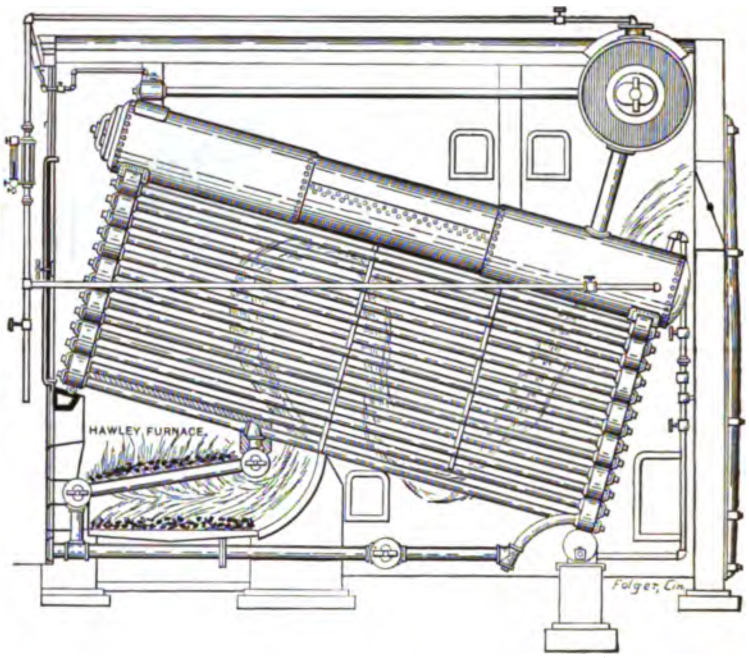


Fig. 141.

CAMPBELL & ZELL WATER TUBE BOILER.

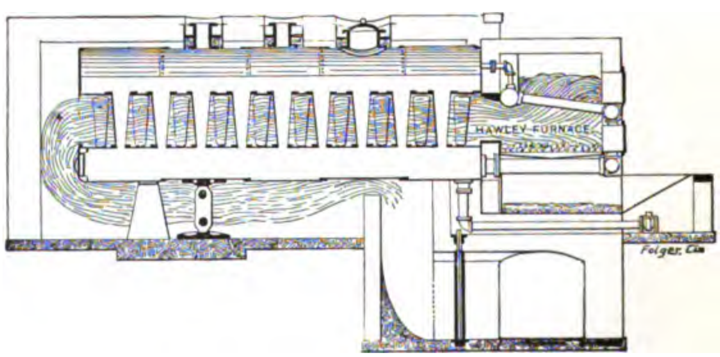


Fig. 142.

GALLOWAY BOILER.

TEMPERATURE OF A FURNACE.

The approximate temperature of a furnace can be obtained in a very simple manner by placing a short, thick piece of iron weighing about 20 pounds in the furnace fire in such a position as to enable it to absorb the greatest heat of the furnace, and allow it to remain a sufficient length of time. Then procure a quantity of water of about three times the weight of the iron. Weigh the iron and water, each, accurately; and just before removing the iron from the furnace take the temperature of the water, then immerse the iron in the water and take the temperature of the water after immersing the iron, and note the highest temperature reached—bearing in mind that the specific heat of iron is about one-ninth that of water—and we are ready.

TO COMPUTE THE HEAT OF A BOILER FURNACE.

RULE.—*Subtract the temperature of the water before immersing the iron in it from the highest temperature after immersing the iron; then multiply the remainder by the number of pounds of water; then multiply the product by the constant whole number* 9, *and call the last product "Product No.* 1." *Next, divide "Product No.* 1" *by the number of pounds contained in the piece of iron, and add the highest temperature of the water to the quotient, the sum will give the temperature of the furnace approximately, or near enough for practical purposes.*

Example.—Let 140° equal temperature of water after immersing heated iron.
Let 45° equal temperature of water before immersing heated iron.
Let 40 pounds equal weight of water.
Let 9° equal specific heat of water, 9 times that of iron.
Let 15 pounds equal weight of iron.

Then we have:

$$\left(\frac{(140-45)\times 40\times 9}{15}\right)+140=2420^\circ \quad \text{Approximate temperature of the furnace.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 140 \\ 45 \\ \hline 95 \\ 40 \\ \hline 3800 \\ 9 \\ \hline 34200 \end{array} \quad \text{" Product No. 1."}$$

Next, dividing "Product No. 1" by 15, the number of pounds contained in the piece of iron, and adding to the quotient 140°, the highest temperature of the water after immersing the iron, we have:

```
15 ) 34200 ( 2280
     30        140
     ---      -----
      42      2420°  Approximate temperature of the
      30                furnace.
     ---
     120
     120
     ---
       0
```

CHAPTER XIII.

SPEED OF PULLEYS AND GEARING.

DIAMETER OF PULLEYS.

TO DETERMINE THE DIAMETER OF A REQUIRED PULLEY.

RULE.—Multiply the number of revolutions per minute of the given pulley by its diameter, in inches, and divide the product by the number of revolutions the required pulley is required to make per minute, and the quotient will give the required diameter of the required pulley in inches.

Example.—Let 175 equal number of revolutions of given pulley per minute.
Let 36 inches equal diameter of given pulley.
Let 420 equal number of revolutions of required pulley per minute.

Then we have:

$$\frac{175 \times 36}{420} = 15 \text{ inches.} \quad \text{Required diameter of required pulley.}$$

Performing the operation, we have:

$$\begin{array}{rrl}
 & 175 & \text{Revolutions of given pulley.} \\
 & 36'' & \text{Diameter of given pulley.} \\
\hline
 & 1050 & \\
 & 525 & \\
\hline
\text{Revolutions of required pulley.}\ 420\,) & 6300\,(& 15 \text{ inches. Required diameter of required pulley.} \\
 & 420 & \\
\hline
 & 2100 & \\
 & 2100 & \\
\hline
\end{array}$$

It will be observed that it makes no difference as to which pulley is the driven pulley or which is the driver, always multiply the number of revolutions per minute of the known pulley by its diameter, in inches, and divide the product by the number of revolutions the other pulley is required to make, and the quotient will give the required diameter of the other pulley in inches.

SPEED OF PULLEYS.

TO DETERMINE THE SPEED OF A REQUIRED PULLEY.

RULE.—Multiply the number of revolutions of the pulley whose speed is known by its diameter in inches, and divide the product by the diameter, in inches, of the pulley whose speed is unknown, and the quotient will give the number of revolutions per minute of the pulley whose speed is unknown.

Example.—Let 175 equal number of revolutions per minute of pulley whose speed is known.

Let 36 inches equal diameter of pulley whose speed is known.

Let 15 inches equal diameter of pulley whose speed is unknown.

Then we have:

$$\frac{175 \times 36}{15} = 420$$ Revolutions of pulley whose speed was unknown.

Performing the operation, we have:

```
                                                 175   Revolutions of pulley whose speed is known.
                                                  36″  Diameter of pulley whose speed is known.
                                                ----
                                                1050
                                                525
                                                ----
Diameter of pulley whose speed is unknown. 15 ) 6300 (420 Revolutions per minute of pulley whose speed was unknown.
                                                60
                                                ---
                                                 30
                                                 30
                                                ---
                                                  0
```

SPEED OF LAST DRIVEN PULLEY IN A SERIES.

TO DETERMINE THE SPEED OF H IN FIG. 143.

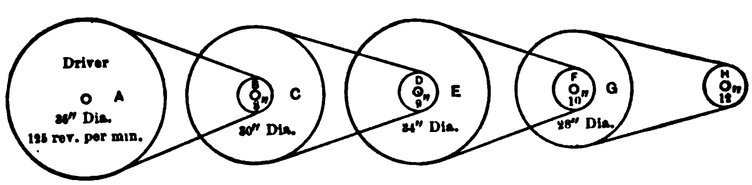


Fig. 143

RULE.—First, multiply the diameters, in inches, of all of the driving pulleys together, A, C, E and G (Fig. 143), and multiply the product by the number of revolutions of the driver A, and call the last product "Product No. 1."

Second, multiply the diameters, in inches, of all of the driven pulleys together, B, D, F and H, and call this product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the number of revolutions per minute of H, the last driven pulley.

Example.—Let 36″ equal diameter of the driver A.
Let 30″ equal diameter of C.
Let 34″ equal diameter of E.
Let 28″ equal diameter of G.
Let 125 equal number of revolutions per minute of the driver A.
Let 8″ equal diameter of driven pulley B.
Let 9″ equal diameter of D.
Let 10″ equal diameter of F.
Let 12″ equal diameter of H.

Then we have:

$$\frac{36\times30\times34\times28\times125}{8\times9\times10\times12}=14875 \text{ Revolutions per minute of driven pulley H.}$$

Performing the operation, we have:

$$\begin{array}{r l}
36'' & \text{A.} \\
30'' & \text{C.} \\
\hline
1080 & \\
34'' & \text{E.} \\
\hline
4320 & \\
3240 & \\
\hline
36720 & \\
28'' & \text{G.} \\
\hline
293760 & \\
73440 & \\
\hline
1028160 & \\
125 & \text{Revolutions of A.} \\
\hline
5140800 & \\
2056320 & \\
1028160 & \\
\hline
128520000 & \text{"Product No. 1."}
\end{array}$$

Multiplying the diameter of the driven pulleys together, we have:

```
   8″  B.
   9″  D.
  ---
  72
  10″  F.
  ---
 720
  12″  H.
 ----
 1440
 720
 ----
 8640  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
8640) 128520000 ( 14875  Revolutions per minute of
      8640                    driven pulley H.
      -----
      42120
      34560
      -----
       75600
       69120
       -----
        64800
        60480
        -----
         43200
         43200
         -----
```

Another method is to determine the number of revolutions of each succeeding pulley separately, thus:

```
                          125     Revolutions of A per minute.
                           36″    Diameter of A.
                          ----
                          750
                         375
                         -----
Diameter of B—     8″) 4500
                         -----
                          562.5    Revolutions of B and C per minute.
                            30″    Diameter of C.
                         -------
Diameter of D—     9″) 16875.0
                         -------
                          1875    Revolutions of D and E per minute.
                            34″   Diameter of E.
                          ----
                          7500
                         5625
                         -----
Diameter of F—    10″) 63750
                         -----
Am't carried forward,     6375    Revolutions of F and G per minute.
```

$$
\begin{array}{rrl}
\textit{Am't brought forward,} & 6375 & \text{Revolutions of F and G per minute.} \\
 & 28'' & \text{Diameter of G.} \\
\hline
 & 51000 & \\
 & 12750 & \\
\hline
\text{Diameter of H— } 12'') & 178500 & \\
\hline
 & 14875 & \text{Revolutions of H per minute.}
\end{array}
$$

SPEED OF LAST DRIVEN PULLEY IN A SERIES.

TO DETERMINE THE SPEED OF A (FIG. 144) WITH H AS A DRIVING PULLEY.

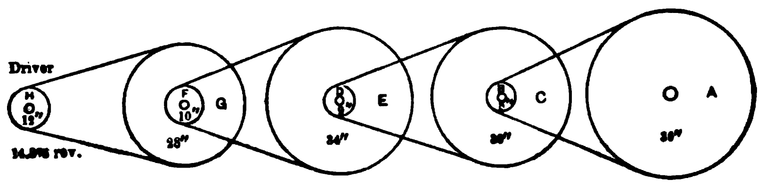


Fig. 144

In this case H, F, D and B are the driving pulleys, and G, E, C and A are the driven pulleys.

RULE.—First, multiply the diameters, in inches, of all of the driving pulleys together, and then multiply the product by the number of revolutions per minute of the driver, which, in this case is H, and call the last product "Product No. 1."

Second, multiply the diameters, in inches, of all the driven pulleys together and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the number of revolutions of the last driven pulley in the series.

Example.—Let 12″ equal diameter of driver H.
Let 10″ equal diameter F.
Let 9″ equal diameter D.
Let 8″ equal diameter B.
Let 14875 revolutions equal speed of driving pulley H.
Let 28″ equal diameter of driven pulley G.
Let 34″ equal diameter E.
Let 30″ equal diameter C.
Let 36″ equal diameter A.

Then we have:

$$\frac{12\times10\times9\times8\times14875}{28\times34\times30\times36}=125 \text{ revolutions. Speed of driven pulley A.}$$

Performing the operation, we have:

```
       12″  Driver.
       10″  Driver.
      ----
      120
        9″  Driver.
     -----
     1080
        8″  Driver.
     -----
     8640
    14875   Revolutions of H.
   ------
    43200
   60480
  69120
 34560
 8640
---------
128520000   "Product No. 1."
```

Next, multiplying the diameter of the driven pulleys together, we have:

```
      28″  Driven.
      34″  Driven.
     ----
     112
     84
     ----
     952
       30″  Driven.
    -----
    28560
      36″   Driven
   ------
   171360
   85680
  -------
  1028160   "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
1028160) 128520000 (125 revolutions.  Speed of driven
         1028160                        pulley A.
         -------
          2570400
          2056320
          -------
           5140800
           5140800
           -------
```

DIAMETER OF LAST DRIVEN PULLEY IN A SERIES.

TO DETERMINE THE DIAMETER OF THE LAST DRIVEN PULLEY.

(FIG. 143).

RULE.—First, multiply the diameters, in inches, of all the driving pulleys together, and multiply the last product by the number of revolutions

per minute of the driver A, and call the product of this operation "Product No. 1."

Second, multiply all the given diameters, in inches, of the driven pulleys together, and multiply the last product by the number of revolutions of the pulley whose diameter it is required to determine, and call the product of this operation "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the required diameter in inches of H, the last driven pulley.

Example.—Let 36 inches equal diameter of driver A.
Let 30 inches equal diameter of C.
Let 34 inches equal diameter of E.
Let 28 inches equal diameter of G.
Let 125 equal number of revolutions per minute of driver A.
Let 8 inches equal diameter of driven pulley B.
Let 9 inches equal diameter of D.
Let 10 inches equal diameter of F.
Let 14875 equal number of revolutions of the last driven pulley H.

Then we have:

$$\frac{36\times30\times34\times28\times125}{8\times9\times10\times14875}=12 \text{ inches.}$$ Required diameter of the last driven pulley H.

Performing the operation, we have:

```
     36"  Diameter of A.
     30"  Diameter of C.
   ----
   1080
     34"  Diameter of E.
   ----
   4320
  3240
  -----
  36720
    28"   Diameter of G.
 ------
 293760
 73440
-------
1028160
    125   Revolutions of A.
-------
 5140800
2056320
1028160
---------
128520000  "Product No. 1."
```

Next, multiplying the known diameters of the driven pulleys together, and multiplying the last product by the revolutions of the pulley whose diameter it is required to determine, and we have:

```
      8″  Diameter of B.
      9″  Diameter of D.
    ---
     72
     10″  Diameter of F.
    ---
    720
  14875   Revolutions of pulley whose speed is
  -----     unknown.
   3600
  5040
 5760
2880
720
--------
10710000  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
10710000) 128520000 ( 12 inches.  Required diameter of the last
          10710000                  driven pulley H.
          ---------
           21420000
           21420000
          ---------
```

DIAMETER OF INTERMEDIATE DRIVING PULLEYS.

TO DETERMINE THE DIAMETER OF THE INTERMEDIATE DRIVING PULLEY E.

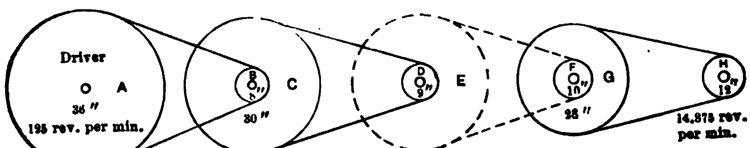


Fig. 145

RULE.—First, multiply the diameters, in inches, of all the driven pulleys together, and multiply the last product by the number of revolutions per minute of the last driven pulley, and call the product of this operation "Product No. 1."

Second, multiply the known diameters, in inches, of all the driving pulleys together, and multiply the last product by the number of revolutions per minute of the first driving pulley, and call the product of this operation "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the required diameter, in inches, of the intermediate driving pulley E.

Example.—Let 8″ equal diameter of first driven pulley B.
Let 9″ equal diameter of D.
Let 10″ equal diameter of F.
Let 12″ equal diameter of H.
Let 14875 equal number of revolutions per minute of last driven pulley H.
Let 36″ equal diameter of first driving pulley A.
Let 30″ equal diameter of C.
Let 28″ equal diameter of G.
Let 125 equal number of revolutions per minute of first driving pulley A.

Then we have:

$$\frac{8\times9\times10\times12\times14875}{36\times30\times28\times125}=34 \text{ inches.}$$ Required diameter of driving pulley F.

Performing the operation, we have:

```
        8″   Diameter of B.
        9″   Diameter of D.
       ---
       72
       10″   Diameter of F.
      ----
      720
       12″   Diameter of H.
     -----
     1440
     720
     -----
     8640
    14875    Revolutions of last driven pulley H.
    -----
    43200
   60480
  69120
 34560
8640
---------
128520000    "Product No. 1."
```

Next, multiplying the known diameters, in inches, of the driving pulleys together, and multiplying the last product by the revolutions per minute of the first driving pulley, A, we have:

```
       36″   Diameter of A.
       30″   Diameter of C.
     ----
     1080
       28″   Diameter of G.
     ----
     8640
    2160
    -----
    30240
```

Am't carried forward, 30240

Am't brought forward, 30240
125 Revolutions of first driving pulley A.

151200
60480
30240

3780000 "Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

3780000) 128520000 (34 inches. Required diameter of driving pulley E.
11340000

15120000
15120000

DIAMETER OF INTERMEDIATE DRIVEN PULLEYS.

TO DETERMINE THE DIAMETER OF INTERMEDIATE DRIVEN PULLEYS.

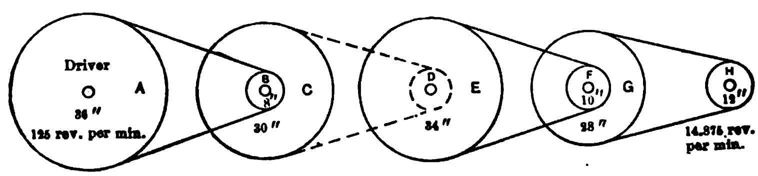


Fig. 146.

RULE.—First, multiply the diameters, in inches, of all the driving pulleys together, and multiply the last product by the number of revolutions per minute of the driver A, and call the product of this operation "Product No. 1."

Second, multiply the known diameters, in inches, of all the driven pulleys together, and multiply the last product by the number of revolutions per minute of the last driven pulley H, and call the last product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the required diameter, in inches, of the intermediate driven pulley D.

Example.—Let 36″ equal diameter of first driving pulley A.
Let 30″ equal diameter of C.
Let 34″ equal diameter of E.
Let 28″ equal diameter of G.
Let 125 equal number of revolutions per minute of first driving pulley A.
Let 8″ equal diameter of first driven pulley B.
Let 10″ equal diameter of F.
Let 12″ equal diameter of H.
Let 14875 equal number of revolutions per minute of last driven pulley H.

Then we have:

$$\frac{36\times30\times34\times28\times125}{8\times10\times12\times14875}=9 \text{ inches.}$$ Required diameter of driven pulley D.

Performing the operation, we have:

36″	Diameter of A.
30″	Diameter of C.
1080	
34″	Diameter of E.
4320	
3240	
36720	
28″	Diameter of G.
293760	
73440	
1028160	
125	Revolutions of first driving pulley A.
5140800	
2056320	
1028160	
128520000	"Product No. 1."

Next, multiplying the known diameter, in inches, of the driven pulleys together, and multiplying the last product by the revolutions per minute of the last driven pulley H, we have:

8″	Diameter of B.
10″	Diameter of F.
80	
12″	Diameter of H.
160	
80	
960	
14875	Revolutions of last driving pulley H.
4800	
6720	
7680	
3840	
960	
14280000	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

14280000) 128520000 (9 inches. Required diameter of driven pulley D
128520000

GENERAL RULE.

In all cases where there is a series of pulleys, the product of the diameters of the complete set of pulleys multiplied by the revolutions per minute of the first driving pulley or last driven pulley, must be divided by the product of the diameters of the incomplete set of pulleys multiplied by the revolutions per minute of the first driving pulley or last driven pulley.

In other words, the complete set of pulleys must always be divided by the incomplete set of pulleys; always performing the operation with the complete set first. If the driving pulleys are complete and the driven pulleys are incomplete, multiply the diameters, in inches, of the driving pulleys and the revolutions per minute of the first driving pulley together, and divide the product by the diameters, in inches, of the driven pulleys and the revolutions per minute of the last driven pulley multiplied together. In all cases containing a series of pulleys the product of the diameters in inches of the driving pulleys multiplied together, must be multiplied by the number of revolutions of the *first* driving pulley; and the product of the diameters, in inches, of the driven pulleys must be multiplied by the *last* driven pulley.

DIAMETERS OF INTERMEDIATE DRIVING AND DRIVEN PULLEYS.

TO DETERMINE THE RESPECTIVE DIAMETERS OF B AND C (FIG. 147).

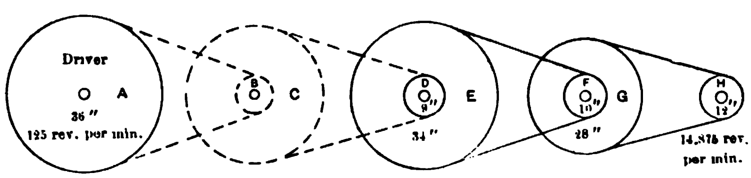


Fig. 147

RULE.—First, multiply the known diameters, in inches, of the given driven pulleys together, and multiply the last product by the number of revolutions per minute of the last driven pulley H, and call the product "Product No. 1."

Second, multiply the known diameters, in inches, of the given driving pulleys together, and multiply the last product by the number of revolutions per minute of the first driving pulley A, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," then assume any convenient diameter for required driven pulley, and multiply the quotient by the assumed diameter, in inches, of the required driven pulley, and the product will give the required diameter, in inches, of the required driving pulley.

Example.—Let 9″ equal diameter of driven pulley D.
Let 10″ equal diameter of F.
Let 12″ equal diameter of H.

Let 14875 equal number of revolutions per minute of H.
Let 36″ equal diameter of driving pulley A.
Let 34″ equal diameter of E
Let 28″ equal diameter of G.
Let 125 equal number of revolutions per minute of A.
Let 8″ equal assumed diameter of the required driven pulley.

Then we have:

$$\left(\frac{9\times10\times12\times14875}{36\times34\times28\times125}\right)\times8=30 \text{ inches.}$$ Required diameter of required driving pulley.

Performing the operation, we have:

```
       9″  Diameter of D.
      10″  Diameter of F.
     ----
      90
      12″  Diameter of H.
     ----
     180
     90
    -----
    1080
   14875  Revolutions of H.
   -----
    5400
   7560
  8640
 4320
1080
--------
16065000  "Product No. 1."
```

Next, multiplying the known diameters, in inches, of the driving pulleys together, and multiplying the last product by the number of revolutions per minute of the first driving pulley A, we have:

```
      36″  Diameter of A.
      34″  Diameter of E.
     ----
     144
    108
    ----
    1224
     28″  Diameter of G.
    ----
    9792
   2448
   -----
   34272
     125  Revolutions of A.
  ------
  171360
  68544
 34272
 -------
 4284000  "Product No 2."
```

Next, dividing "Product No. 1" by "Product No. 2," we have:

```
4284000) 16065000 (3.75   The Quotient.
         12852000
         --------
          32130000
          29988000
          --------
           21420000
           21420000
           --------
```

Finally, multiplying the quotient by the assumed diameter of the required driven pulley, we have:

```
 3.75
    8"   Assumed diameter of required driven pulley B.
 -----
30.00 inches. Required diameter of required driving pulley C.
```

DIAMETERS OF DRIVING AND DRIVEN PULLEYS ON INTERMEDIATE SHAFT TO MAKE A GIVEN NUMBER OF REVOLUTIONS.

TO DETERMINE THE DIAMETERS OF DRIVEN AND DRIVING PULLEYS FOR INTERMEDIATE SHAFT REQUIRED TO MAKE A GIVEN NUMBER OF REVOLUTIONS PER MINUTE (FIG. 148).

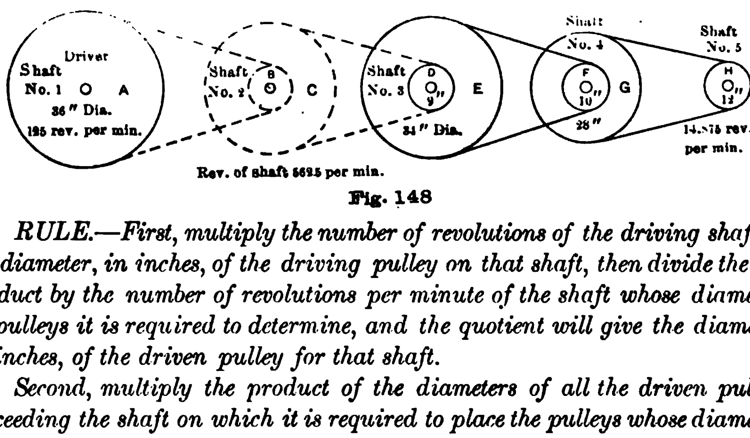


Fig. 148

RULE.—First, multiply the number of revolutions of the driving shaft by the diameter, in inches, of the driving pulley on that shaft, then divide the last product by the number of revolutions per minute of the shaft whose diameters of pulleys it is required to determine, and the quotient will give the diameter, in inches, of the driven pulley for that shaft.

Second, multiply the product of the diameters of all the driven pulleys succeeding the shaft on which it is required to place the pulleys whose diameters it is required to determine, and multiply the product by the number of revolutions per minute of the last driven pulley, and call the last product "Product No. 1."

Third, multiply the product of the diameters of all the driving pulleys succeeding the shaft on which it is required to place the pulleys whose diameters it is required to determine, and multiply the product by the number of revolutions of the shaft on which the required pulleys are to be placed, and call the last product "Product No. 2."

Fourth, divide "Product No. 1" by "Product No. 2," and the quotient will give the required diameter, in inches, of the driving pulley for the shaft on which the required pulleys are to be placed.

Applying the rule to a series of driving and driven pulleys, as shown in Fig. 148, proceed as follows to determine the required diameter of the driven pulley B on shaft No. 2:

RULE.—Multiply the number of revolutions per minute of shaft No. 1 (the driving shaft) by the diameter, in inches, of driving pulley A on that shaft, and divide the product by the number of revolutions per minute shaft No. 2 is required to make, and the quotient will give the required diameter of the driven pulley B on that shaft.

Example.—Let 125 equal number of revolutions per minute of shaft No. 1.
Let 36 inches equal diameter of driving pulley A.
Let 562.5 equal number of revolutions per minute shaft No. 2 is required to make.

Then we have:

$$\frac{125 \times 36}{562.5} = 8 \text{ inches.}$$ Required diameter of driven pulley B on shaft No. 2.

Performing the operation, we have:

	125	Revolutions of shaft No. 1 per minute.
	36″	Diameter of driving pulley A.
	750	
	375	
Revolutions of shaft No. 2 per minute. 562.5)	4500.0 (8 inches.	Required diameter of driven pulley B on shaft No. 2.
	4500.0	

Next, in determining the diameter of the driving pulley C on shaft No. 2 (Fig. 148), proceed as follows:

RULE.—First, multiply the diameter of all of the driven pulleys together, D, F, and H, located to the right in the engraving, and multiply the last product by the number of revolutions per minute the last driven pulley H is required to make, and call the last product "Product No. 1."

Second, multiply the diameters of the driving pulleys together, E and G, and multiply the product by the required number of revolutions per minute shaft No. 2 is required to make, and call the last product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the required diameter of driving pulley C on shaft No. 2.

Example.—Let 9″ equal diameter of driven pulley D.
Let 10″ equal diameter of F.
Let 12″ equal diameter of H.
Let 14875 equal revolutions per minute of H.
Let 34″ inches equal diameter of driving pulley E.
Let 28″ inches equal diameter of G.
Let 562.5 equal revolutions per minute of shaft No. 2.

Then we have:

$$\frac{9\times10\times12\times14875}{34\times28\times562.5}=30 \text{ inches.}$$ Required diameter of driving pulley C on shaft No. 2

Performing the operation, we have:

```
       9″   Diameter of D.
      10″   Diameter of F.
     ---
      90
      12″   Diameter of H.
     ---
     180
      90
    ----
    1080
   14875    Revolutions of shaft No. 4.
   -----
    5400
   7560
  8640
 4320
1080
--------
16065000    "Product No. 1."
```

Next we have:

```
  34″   Diameter of E.
  28″   Diameter of G.
 ---
 272
 68
 ---
 952
 562.5  Revolutions of shaft No. 2.
 -----
  4760
 1904
5712
4760
--------
535500.0  "Product No. 2."
```

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
535500 ) 16065000 ( 30 inches.   Required diameter of driving pulley C on shaft No. 2.
         1606500
         -------
               0
```

SPEED OF GEARING.

The rules relating to the speed of pulleys relate also to the speed of gearing with one notable exception, which is explained further on. The diameter of the gear wheels must be taken at the pitch line when diameters are employed in making calculations. The number of teeth, however, may be taken instead of diameters. The exception in the

rule mentioned above occurs when the driving and driven shafts are to be placed a given distance apart. With this single exception, the rules laid down for determining the speed of pulleys are applicable in determining the speed of gearing. Hence, it is unnecessary to reiterate them here.

DIAMETER OF GEAR WHEELS.

TO DETERMINE THE REQUIRED DIAMETER OF THE DRIVING WHEEL.

RULE.—First, multiply the distance, in inches, the shafts are apart from center to center, on which the wheels are to run, by 2, and call the product "Product No. 1."

Second, add the number of revolutions per minute of the driving and driven wheels together, and divide the sum by the number of revolutions per minute of the driven wheel, and call the quotient "Quotient No. 1."

Third, divide "Product No. 1" by "Quotient No. 1," and the quotient of this operation will give the diameter of the pitch line of the driving wheel in inches.

Example.—Let 18 inches equal distance required between centers of wheels.
Let 2 equal a constant.
Let 150 equal revolutions of driving wheel per minute.
Let 30 equal revolutions of driven wheel per minute.

Then we have:

$$\frac{18\times2}{(150+30)\div30}=6 \text{ inches.}$$ Required diameter of driving wheel at the pitch line.

Performing the operation, we have:

$$\begin{array}{rl} 18 & \text{Distance between centers.} \\ 2 & \text{A constant.} \\ \hline 36 & \text{"Product No. 1." This product represents the combined diameter of both wheels.} \end{array}$$

Next, adding the number of revolutions of both wheels together and dividing the sum by the revolutions of the driving wheel, we have:

$$\begin{array}{rl} 150 & \text{Revolutions of driving wheel.} \\ 30 & \text{Revolutions of driven wheel.} \\ \hline 30)\,180\,(6 & \text{"Quotient No. 1."} \\ 180 & \\ \hline \end{array}$$

Finally, dividing "Product No. 1" by "Quotient No. 1," we have:

$$\begin{array}{rl} 6)\,36 & \\ \hline 6 \text{ inches.} & \text{Required diameter of the driving wheel at the pitch line.} \end{array}$$

TO DETERMINE THE REQUIRED DIAMETER OF THE DRIVEN WHEEL.

RULE.—First, multiply the distance, in inches, the wheels are to run apart, from center to center, by 2, and call the product "Product No. 1."

Second, add the number of revolutions per minute of the driving and driven wheels together, and divide the sum by the number of revolutions per minute of the driving wheel, and call the quotient "Quotient No. 1."

Third, divide "Product No. 1" by "Quotient No. 1," and the quotient of this operation will give the diameter of the driven wheel at the pitch line in inches.

Example.—Let 18 inches equal distance required between centers of wheels.

Let 2 equal a constant.

Let 150 equal revolutions of driving wheel per minute.

Let 30 equal revolutions of driven wheel per minute.

Then we have:

$$\frac{18\times 2}{(150+30)\div 150}=30 \text{ inches.}$$ Required diameter of driven wheel at the pitch line.

Performing the operation, we have:

18	Distance between centers.
2	A constant.
36	"Product No. 1." This product represents the combined diameter of both wheels.

Next, adding the number of revolutions of both wheels together, and dividing the sum by the revolutions of the driven wheel, we have:

```
      150  Revolutions of driving wheel.
       30  Revolutions of driven wheel.
      ---
150)  180 (1.2  "Quotient No. 1."
      150
      ---
       300
       300
       ---
```

Finally, dividing "Product No. 1" by "Quotient No. 1," we have:

```
1.2) 36.0 (30 inches.  Required diameter of the driven
     36                wheel at the pitch line.
     --
      0
```

It will be noticed that there is very little difference in the form of the above examples, and they are therefore here given for comparison.

To determine diameter of driving wheel:

$$\frac{18\times 2}{(150+30)\div 30}=6 \text{ inches.}$$

To determine diameter of driven wheel:

$$\frac{18\times 2}{(150+30)\div 150}=30 \text{ inches.}$$

It will be observed that the only difference is in the divisors, by which the sum below the line is divided. In the first case, in order to determine the required diameter of the driving wheel, we divide by the revolutions of the driven wheel. In the second case, in order to determine the required diameter of the driven wheel, we divide by the revolutions of the driving wheel. Therefore, in all cases where the diameter of one wheel is required, we must divide the sum below the line by the revolutions of the other wheel. However, but one of the above operations is necessary for determining the required diameter of both wheels. After the diameter of one of the wheels is determined—and it may be either—the diameter of the other may be determined as follows:

RULE.—*Multiply the number of revolutions of the wheel whose diameter has been determined, according to any of the foregoing rules, by the diameter of such wheel, and divide the product by the number of revolutions of the other wheel, and the quotient will give the required diameter of the other wheel.*

Example.—Let 6 inches equal the ascertained diameter of one of the wheels.

Let 150 equal revolutions of wheel whose diameter has been ascertained.

Let 30 equal revolutions of wheel whose diameter is to be determined.

Then we have:

$$\frac{150\times 6}{30}=30 \text{ inches.}$$ Required diameter of the other wheel.

Performing the operation, we have:

$$\begin{array}{r l} & 150 \\ & \underline{\quad 6} \\ 30) & 900\ (30 \text{ inches.} \\ & \underline{90} \\ & \quad 0 \end{array}$$ Required diameter of the other wheel.

On the other hand:

Let 30 inches equal diameter of driven wheel.
Let 30 equal revolutions per minute of driven wheel.
Let 150 equal revolutions of driving wheel.

Then we have:

$$\frac{30\times 30}{150}=6 \text{ inches.}$$ Required diameter of driving wheel.

Performing the operation, we have:

```
          30
          30
        ----
    150 ) 900 (6 inches.   Required diameter of driving wheel.
          900
          ---
```

DIAMETER OF GEAR WHEELS—SIMPLE METHOD.

TO DETERMINE THE REQUIRED DIAMETER OF THE DRIVING WHEEL.

RULE.—First, multiply the distance, in inches, between the centers of the shafts by 2, and multiply the product by the number of revolutions of the driven wheel per minute, and call the last product "Product No. 1."

Second, add the number of revolutions per minute of both wheels together and call the total "The Sum."

Third, divide "Product No. 1" by "The Sum," and the quotient will give the required diameter of the driving wheel in inches.

Example.—Let 18 inches equal distance between centers of the shafts.

Let 2 equal a constant.

Let 30 equal number of revolutions per minute of driven wheel.

Let 150 equal number of revolutions per minute of driving wheel.

Then we have:

$$\frac{18 \times 2 \times 30}{150+30} = 6 \text{ inches.}$$ Required diameter of driving wheel.

Performing the operation, we have:

```
   18"   Distance between centers of shafts.
    2    A constant.
  ----
   36"   Combined diameter of both wheels.
   30    Revolutions per minute of driven wheel.
  ----
 1080    "Product No. 1."
```

Next we have:

```
  150    Revolutions per minute of driving wheel.
   30    Revolutions per minute of driven wheel.
  ----
  180    "The Sum."
```

Finally, dividing "Product No. 1" by "The Sum," we have:

```
 180 ) 1080 ( 6 inches.   Required diameter of driving wheel.
       1080
       ----
```

TO DETERMINE THE REQUIRED DIAMETER OF THE DRIVEN WHEEL.

RULE.—First, multiply the distance, in inches, between the centers of the shafts by 2, and multiply the product by the number of revolutions of the driving wheel per minute, and call the product "Product No. 1."

Second, add the number of revolutions per minute of both wheels together, and call the answer "The Sum."

Third, divide "Product No. 1" by "The Sum," and the quotient will give the required diameter of the driven wheel in inches.

Example.—Let 18 inches equal distance between centers of the shafts.
Let 2 equal a constant.
Let 150 equal number of revolutions per minute of the driving wheel.
Let 30 equal number of revolutions per minute of the driven wheel.

Then we have:

$$\frac{18\times2\times150}{150+30}=30 \text{ inches. Required diameter of driven wheel.}$$

Performing the operation, we have:

18″	Distance between centers of shafts.
2	A constant.
36″	Combined diameter of both wheels.
150	Revolutions per minute of driving wheel.
1800	
36	
5400	"Product No. 1."

Next we have:

150	Revolutions per minute of driving wheel.
30	Revolutions per minute of driven wheel.
180	"The Sum."

Finally, dividing "Product No. 1" by "The Sum," we have:

180) 5400 (30 inches. Required diameter of driven wheel.
540
0

CHAPTER XIV.

ELECTRICITY.

The great field now occupied by this comparatively newly developed science makes it imperative for steam engineers to study and understand it, or be relegated to the lower ranks of steam engineering. In order, then, to afford an opportunity to the engineer and student of steam engineering to become acquainted with some of the rudiments and elementary principles of this important science, so as to better prepare them to engage in the study of the higher and more intricate branches of electrical engineering, this chapter will be devoted to a short and simple elucidation of a few of the more plainer principles upon which this science has been founded.

The first and most important step for the student to take is to thoroughly acquaint himself with the meaning of the terms employed in connection with this branch of steam engineering, as well as with the laws governing the measurement of resistance, pressure, quantity and power developed by electricity.

DEFINITION OF TERMS—ELECTRICAL UNITS.

The following resolutions were adopted by the Electrical Congress at Paris, in 1881:

First. In electrical measurements the three fundamental units shall be adopted: centimeter, gramme and second (c. g. s.).

Second. The practical unit, the ohm and the volt, will preserve their actual value: 10^9 for the ohm, and 10^8 for the volt.

Third. The unit of resistance, the ohm, will be represented by a column of mercury of one square millimetre section at the temperature zero centigrade—the length of this column is approximately 1.05 metres.

Fourth. The current produced by a volt through an ohm shall be called an ampere.

Fifth. The unit of quantity, the coulomb, shall be the quantity of electricity defined by the condition that an ampere gives a coulomb per second.

Sixth. A farad, the unit of capacity, shall be the capacity defined by the condition that a coulomb in a farad gives a volt.

ELECTRICAL RESISTANCE.

Electrical resistance is that property of conductors by virtue of which they tend to reduce the intensity of a current passing through them. The practical unit of resistance is the ohm, which, according to Ohm's Law, may be stated as follows: The strength of the current varies directly as the electro-motive force, and inversely as the total resistance.

Or, adopting the units employed by practical electricians, it may be stated in this way: The number of amperes of current flowing through a circuit is equal to the number of volts of electro-motive force divided by the number of ohms of resistance in the entire circuit.

First. The resistance of a conducting wire is directly proportional to its length.

Second. The resistance of a conducting wire is inversely proportional to the area of its cross section, and therefore in the usual round wires is inversely proportional to the square of its diameter:

Third. The resistance of a wire of a given length and given thickness depends upon the material of which it is made.

ELECTRO-MOTIVE FORCE.

Electro-motive force is the name given to that which moves or tends to move electricity from one place to another. For brevity, it is sometimes written E. M. F. The practical E. M. F. unit is called the volt.

A mechanical horse power is 33,000 pounds lifted the height of one foot in one minute's time.

An electrical horse power is 746 watts. A watt is one ampere multiplied by one volt.

A DYNAMO.

NAMES OF THE DIFFERENT PARTS.

The names of the different parts of the dynamo, as shown in Fig. 149, are as follows:

A A, field magnets; B, armature; C, commutator; D D, bearings; E, pulley; F, switch; G G, brushes and brush holder; H H, binding posts; I I, oil cups; J, base frame and yoke; K K, pole pieces.

These parts, assembled together in one machine, comprise a dynamo or motor.

HOW TO BUILD A DYNAMO.

TO PRODUCE A TWELVE VOLT MACHINE.

First, procure a complete set of castings, then drill, tap and fit all the parts neatly and accurately; then assemble the field magnets by

securing the two castings in an upright position to the base frame, with large, square-headed bolts going through the bottom of the base into the field castings.

Next get the armature shell ready. Saw twenty-four notches, about one-eighth of an inch deep, in the edge of flanges in line with one another. In these notches insert thin pieces of fibre the width of the flanges, and about one-eighth of an inch projection above the edge of the flanges.

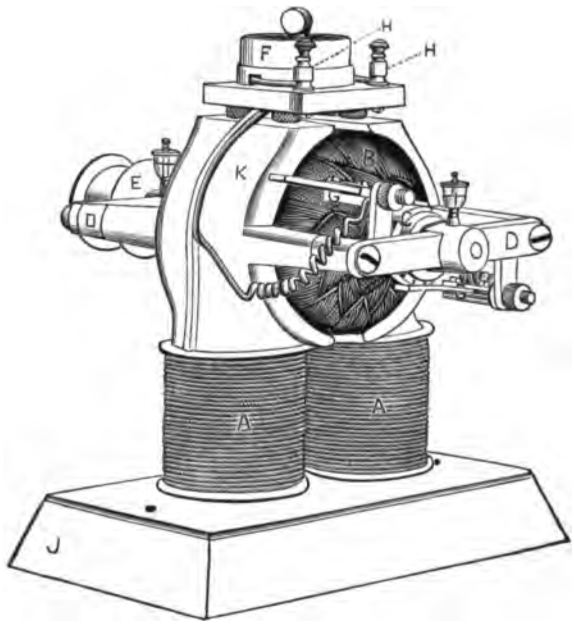


Fig. 149.

Then wind soft iron wire, of about No. 12 gauge, on the armature shell, first soldering the inner end of the wire to the iron. Wind the shell full until the wire comes up to the outer edges of the flange, keeping the surface of the wire as level as possible. Solder the outer end of the wire to the outer surface of the wire to keep it in place.

Now insulate the entire surfaces of the iron armature core with about two layers of thin cloth, coating the cloth with shellac or glueing the edges down to make it stick in place. The ends of the armature core and a portion of the shaft, about three-fourths of an inch from the ends of the armature core, should also be insulated. This insulation is placed on to keep the wire from coming in metallic contact with any portion of the frame.

TO WIND THE ARMATURE.

After having the armature core thoroughly insulated, the next thing to be done is to wind the armature core with about one pound of No. 18, B. & S. gauge, double cotton-covered wire. Start the wire on top of armature between space divided by fibre projections, allowing the end of the wire to project at least 6 inches. The end can be drawn around one of the fibre projections in order to keep it in place.

Start the wire on the left side of the space, and carry it by one hand along the surface of the armature core and through the space between the two fibre wedges at the opposite end of the core corresponding with the space in which the coil is started. With the other hand grasp the armature core, and as the wire is carried across the end of the armature core, the armature is turned through a half revolution, bringing the opposite side of the core uppermost. The wire is then laid between the two pieces of wedges diametrically opposite those embracing the wire. On the opposite side of the armature the wire is carried across the commutator end of the armature core, and the armature is returned to the position of starting by returning to its first position, and the wire is laid alongside of the portion first laid on The wire is carried lengthwise around the armature in this manner until the space between the fibre projections is filled with one layer. This will bring eight turns around the entire armature to each layer.

In winding, the wire should be kept in equal proportions on the opposite side of the shaft, in order to keep the armature in balance.

After one layer of wire has been put on in this manner, the operation must be continued until four layers have been put on, which will make four wires deep and eight wires wide, which constitutes one coil.

Bring the two ends out, allowing at least 6 inches on each wire, and cut the wire off the spool, and to that end attach a card marking it "*outside end of first coil.*" The two ends may be twisted together, so as to hold them in place.

Now proceed to wind on the next coil. Revolve the armature half way and commence the coil alongside the first on the left side at the opposite side of the armature where the ends of the first coil come out. Wind this in the same manner as the first, or No. 1 coil, was wound; bring the ends out on the left-hand side and attach a card on the outside end, marking it "*outside end of second coil.*"

Next, proceed to wind coil No. 3. Turn the armature half-way and again commence on the left-hand side of coil No. 1. Wind in the same manner in which Nos. 1 and 2 were wound, and bring the ends out on the same side of the armature No. 1 commenced on. This end must be marked "*outside end of third coil.*"

Next, begin to wind coil No. 4. Turn the armature over, and begin in the same manner in which Nos. 1, 2, and 3 coils were begun. Begin on the left-hand side in space, wind the same as the others, bring the last end out, cut it off and twist ends together, marking the outside end, "*outside end of fourth coil.*"

It must be borne in mind that each coil should have eight complete turns and four complete layers, making the wind on the armature four layers deep and eight layers wide.

In winding the coils, care must be taken to keep the wires divided on equal sides of the shaft at the end, in order to keep the armature in electrical and mechanical balance.

TO HANDLE THE ARMATURE.

When winding the armature, take a small box, saw notches in the ends and rest the shafts in the notches, so the armature core can be turned to bring it in the necessary position while winding.

After the armature is all wound, tie two cords on the center of the armature, about 1¼ inches apart, under these cords slip a thin piece of mica or cloth—mica is the best; then put five or six narrow strips of thin copper under these cords and over the mica. The cord is employed to hold the mica bands in position.

The next thing to be done is to band the armature. This is done by wrapping fine brass wire over the top of the mica, and continuing until the wrapped wire has reached a width of 1 inch, or 1¼ inches. The little cleats of thin copper under the brass wire must then be bent up and over the top of the brass wire, and hammered down slightly. Then solder the whole thing; that is, solder the brass wire together, so as to have a firm band of wire about 1¼ inches wide. It should be well insulated underneath with mica or cloth, to keep the band from coming in contact with the armature wire.

TO MAKE AND CONNECT THE COMMUTATOR.

First, take the commutator shell, the piece that goes on the end that clamps the commutator segments in place, true up and bevel the inside, so as to fit the commutator segments before the segments are sawed out. The end of the commutator shell opposite the one that has the shoulder extending to it, should be tapped, and the hexagon nut should be bored and tapped to screw into the place which is used to bind the segments together; then drill a hole in the center of the shell for the shaft.

After these parts are all fitted, separate the segments by running a saw in the slots. There will be, when separated, twelve pieces, each of which should be numbered, so as to assemble side by side in their required positions.

Before placing segments on the shell it will be necessary to insert a narrow piece of mica between each segment. The mica should be about one-thirty-second of an inch thick.

Take a small piece of wood, or other material, and assemble the segments around it in their proper order; then place a soft rubber band around the segments to hold them in a circle in their proper position; then insert a slip of mica between each segment, so that they are all thoroughly insulated and do not come in metallic contact with one another. Then take the diameter of the segments as they are, and take two pieces of wood and halve them out so as to encircle the entire commutator; clamp the two pieces of wood around the segments by putting the whole into a vise, and drawing them tightly together. Then bind the two ends of the wood tightly together with two pieces of wire. The wood is used to handle the segments after they are in their proper position, and should not exceed three-eighths of an inch in thickness. Next, insulate the core of the commutator shell with thick cloth; then insulate each end of the shell on the inside with thick cloth, which is done to keep the segments from coming in metallic contact with the outer portion of the shell; in short, each segment must be insulated from its neighbor and also from the commutator shell. Next, make a small key-seat in the commutator where it goes on the shaft, then drill a hole in the shaft and insert a small metal point to serve as a key. This is done to hold the commutator in proper position after it is mounted to the armature shaft. Before placing the commutator on the shaft, it should be placed in a lathe and the entire surface trued up. After the commutator is mounted the wires should be soldered in their respective segments; there should be a groove filed in each segment at the shoulder, the wires embedded in the grooves and then soldered. Care must be taken to get good metallic contact in each case.

TO CONNECT THE ARMATURE.

Connect the outside end of the first coil to the inside end of the alternate coil on the right-hand side; then the outside of the last named coil to the inside of the next alternate coil on the right-hand side, and so on until the twelve coils are connected to their respective segments in the commutator, giving the commutator segments what is termed a "lead" of one segment.

TO WIND THE FIELD MAGNETS.

Cut two fibre heads for each magnet, 3¼ inches in diameter, with a hole in each, sufficiently large to allow these heads to slip over the cast-iron field core and fit tight. Slip one head to each end of the field core, and insulate the space between them with thin cloth; then drill a small hole in one of the heads next to the pole piece end for the

inner wire to project through. Put the inner wire through this hole, and insulate the portion of wire projecting through the head with a small rubber tube. Next wind the field coil full—eight layers deep in all—with No. 16 B. & S. double cotton-covered wire. Next drill a hole in the opposite side of the same fibre head that the first wire went through. This hole should be near the outer edge of fibre head, and the last end of the wire should be brought through this hole. The outer field magnet should be wound exactly like the first.

The field should be unscrewed from the base when ready for winding, and replaced when both are wound, after which connect the two inside ends together. Care must be taken to get good metallic contact. One of the outside terminals must be connected to the brush holder, and the other brush holder must be connected to one binding post, and the other outside field connection must be connected to the opposite binding post.

After winding the fields and armature, they should be well coated, when finished, with shellac varnish. This will prevent them from absorbing moisture, and also will insulate and tend to preserve them.

TO INSULATE THE BRUSH HOLDERS.

The brush holders must be insulated from the frame, with paper, rubber or fibre washer, placed one on each side of the frame. There should also be a small fibre bushing fitted into the frame where the brush holder is attached. In short, all parts of the machine in circuit with the wires should be thoroughly insulated.

TO MAKE THE BRUSHES.

Take about fourteen strips of No. 30 spring copper, three-eighths of an inch wide and three inches long. One end of these should be soldered together so as to make a brush that will give sufficient elasticity. To clamp the brushes, take a thin piece of brass the width of the brush holder, drill one hole in each end and attach it to the brush holder with two machine screws, which will serve as a clamp for holding the brushes in position.

TO OPERATE THE MACHINE.

First run the machine as a motor, or send a current through the fields to charge them. Before an electric current can be generated by rotating the armature, the field magnets must contain some magnetism, no matter how small an amount. This is known as residual magnetism, and it must be apparent in a machine before it is possible to generate electricity by cutting the lines of force at right angles in rotating the armature. When the armature is rotated, a feeble current is set up in it, and passes through the field magnets, fully exciting them, and

as the armature continues to rotate the current is built up until the machine gives out its normal capacity.

Set the upper brush and shift it backward and forward until the spark is cut down to the lowest possible minimum. The brushes should not bear too hard on the commutator, but still have sufficient pressure to insure a good electrical contact. It will be well to use a little light grade of oil on the commutator occasionally, such as sperm oil; but it will be found better to use a heavier grade of oil on the bearings.

This machine, when run as a dynamo, will operate ten 6 candle-power lamps, at 3500 revolutions per minute, when loaded at 12 volts pressure, giving a quantity of 15.55 amperes. When run as a motor it will require 12 volts pressure to develop its full power, which will be a trifle less than one-fourth horse power. Five cells of battery will run it very well; still it will require eight to bring it up to its full power.

50 VOLTS, E. M. F.

This machine can be wound for 50 volts potential. The winding is done in precisely the same manner as described for a 12 volt machine, with the exception that there must be twenty-six layers of No. 24 wire on the fields, and six layers of No. 22 wire on the armature. Eight turns to each layer.

110 VOLTS, E. M. F.

The coupling of the armature is precisely the same as in the 12 volt machine, with the exception that the 12 volt machine is coupled in what is known as "series" winding, while the 50 and 110 volt machines are coupled in what is known as "shunt" winding. In the 50 and 110 volt machines, the two inner ends of the field coils are connected together. The two outside ends are connected to the two binding posts, H H, as shown in Fig. 149. Each of the two outside ends is connected to its respective binding post separately. The two wires leading to the brushes, G G, should also be connected to the binding posts, H H, one wire to each post, bringing the outside of one field coil, A, and the wire from one brush holder, G, together at one post, H, on the left-hand side, and bringing the outside wire of the other field coil A, to the wire leading from the brush holder on the right-hand side to the binding post H, on the right-hand side.

The two wires that lead from the machine to lamps are connected to the binding posts, H H. The switch, F, must connect in the circuit anywhere, so as to open the circuit or close it when the current is to be turned off.

This machine is known as "shunt wound," and at 3500 revolutions per minute, will give an electro-motive force of 50 volts at 5

amperes, or 110 volts at 3 amperes, with a capacity to operate five 16 candle-power (50 or 110 volts) lamps, requiring about one-half horse power. Or it can be operated as a one-fourth horse power motor when connected in a 50 or a 110 volt circuit.

MULTIPLE SYSTEM OF COUPLING.

To connect the lamps to this machine, either 12, 50 or 110 volts, stretch two wires from the respective binding posts at any desired point, keeping the wires separate. The wire leading from one side of the machine is known as the positive or outgoing wire, and the wire returning on the other side of the machine is known as the negative or returning wire. To connect lamps one wire of the lamp should be connected to the positive or outgoing wire, and the other wire of the lamp should be connected to the negative or returning wire. The lamps can be connected in this manner until all the lamps in the circuit that the machine will carry are connected. This manner of coupling lamps in the circuit is known as "multiple coupling."

To burn lamps of less voltage they can be connected in what is known as "series." The current goes from lamp No. 1 to No. 2, from No. 2 to No. 3, and so on; the last wire to the last lamp being connected to the negative or wire returning to the machine.

In the series method of coupling, five lamps requiring 10 volts each, coupled in series, would be equal to one lamp of 50 volts current in multiple. In the series coupling, the entire current of the machine passes through each lamp; the lamps being all in direct series with the machine. In the multiple coupling, each lamp takes only its portion of the current; that is to say, if two lamps are coupled in multiple, the current is split twice; if five lamps, the current is split five times; and so on.

CHAPTER XV.

THE STEAM ENGINE INDICATOR.

The steam engine indicator is employed as an instrument of communication with the interior of the steam engine while in operation. By its instrumentality we are enabled to diagnose the condition of the various organs of the engine's interior, with even more accuracy than a skillful physician can the condition of the various organs of the interior of the human body.

It not unfrequently happens that even the most skillful physicians are unable to determine the cause, nature, or location of a disease, except by a *post mortem* examination. While, with the indicator in the hands of a skillful engineer, the cause, nature and location of any disease of the steam engine's interior can be determined with even more accuracy than can be done by any ocular inspection. Any defect in the construction, movement, or setting of the valves, can be easily and readily detected. Any defect or deficiency can be located and determined with the utmost accuracy and precision; and hence the skillful engineer will be able to prescribe, with certainty, a remedy that will cure or correct any defect or deficiency in the steam engine pointed out on the diagram of the steam engine indicator.

The indicator not only points out and locates any defects in the interior working parts of the steam engine, but it gives the quality, quantity, and temperature of steam in the cylinder; the mechanical force exerted by it during every part of the piston's motion, and the quantity of water consumed in performing any given amount of work.

The transmission of all this information to the engineer by the indicator is not to satisfy mere idle curiosity, but for the important purpose of determining the economy of the steam engine; for, after all, the engine that will do the greatest amount of work at the least expense is undoubtedly the best engine. It is, therefore, of the utmost importance that the working parts of each engine should be so proportioned and so adjusted as to produce the best possible results.

In this question of economy the whole construction of the steam engine is involved; and to determine the question with accuracy it is necessary to ascertain just what is going on in the interior of the engine when it is in operation, as it can be determined in no other way; and from the information recorded by the indicator the engi-

neer is enabled to determine the nature and location of defects and to apply the remedy, in such a way that the best possible results, according to the principle upon which the engine is constructed, may be produced.

The qualifying remark is made for the reason that there is a vast difference in the construction of the various types of steam engines, and no adjustment of their working parts will place them exactly on the same level in point of economy; but what can be done with the indicator is to enable the engineer to adjust and proportion the valves of each particular type of engine, so as to produce the best possible results for that particular engine, so far as the principle of its construction will admit. It can not, therefore, be too strongly urged upon the engineer, as well as upon the student of steam engineering, to become thoroughly familiar with the use of the steam engine indicator.

DESCRIPTION OF THE INDICATOR.

In imparting the information relating to the construction of the indicator, the Tabor indicator has been selected for illustration. Not because of its real or supposed superiority, nor to give it any advantage, over other first-class instruments; but because: First, for the lack of space to give a description of each particular make; and, second, because it embodies all of the essential features necessary to perform the work required of such instruments. It is, therefore, no disparagement whatever to any other first-class indicator that it, instead of any other make, has been selected. The student will, therefore, understand that in studying this description he is simply studying the construction of all first-class indicators and not any particular make.

One of the features of the indicator here described is the means employed to communicate a straight line movement to the pencil, as shown in Figs. 150, 151 and 152.

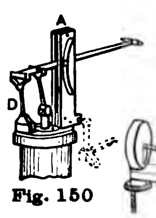


Fig. 150

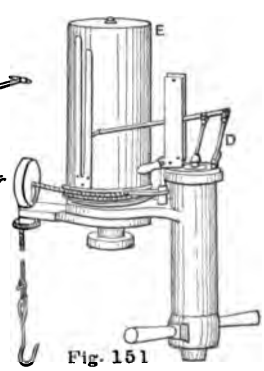


Fig. 151

FRONT VIEW OF STANDARD INSTRUMENT AND BACK VIEW OF PENCIL MECHANISM.

The stationary plate A, Fig. 150, and B, Fig. 152, is attached, in an upright position, to the cover of the steam cylinder. This plate contains a curved slot, which serves as a guide and controls the motion of the pencil bar. The side of the pencil bar is provided with a roller

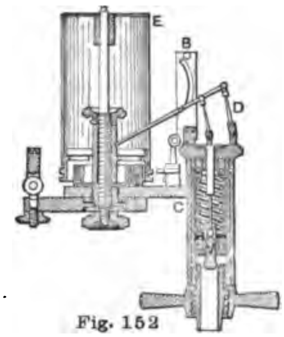


Fig. 152

SECTIONAL VIEW OF STANDARD INSTRUMENT.

which turns on a pin, and fitted so as to roll freely from end to end of the slot with little lost motion. The radius of the curve in the slot is such, and the pin carrying the roller is attached to such a point in the pencil bar, that the end of the pencil bar which carries the pencil, moves up and down in a straight line, when the roller is moved from one end of the slot to the other.

The steam cylinder of this indicator and the base of the paper drum are made in one casting. Inside of the steam cylinder C is a movable lining within which the piston of the indicator works. This inside cylinder is attached by means of a screw thread at the bottom, and openings on the opposite sides at the tops are provided for the insertion of a tool for screwing it in or out. Openings through the sides of the outer cylinder are provided to allow the steam which leaks by the piston to escape. The pencil mechanism is carried by the cover of the outside cylinder. The cover proper is stationary, but a swivel plate, which extends over nearly the whole of the cover, has the pencil mechanism attached to it. By means of this swivel plate, the pencil bar may be turned into contact with the paper drum, as is done when taking a diagram. The pencil mechanism is attached to the swivel by means of the vertical plate containing the slot, and a small standard placed on the opposite side of the swivel for connecting the back link D (Figs. 150, 151 and 152). The slotted plate is backed by a plate of similar size, which serves to receive the pressure brought to bear on the pencil when taking diagrams, and to keep the pencil bar in place. The pencil mechanism consists of three pieces—the pencil bar, the piston-rod link and the back link. The two links are parallel with each other during every part of the stroke of the pencil bar. The

connection between the piston and the pencil mechanism is made by means of a steel piston rod. At the upper end where it passes through the cover it is hollow, and has an outside diameter of three-sixteenths of an inch; at the lower end it is solid and its diameter is reduced. It connects with the piston through a ball and socket joint. The socket forms an independent piece, which fits into a square hole in the center of the piston, and is fastened by means of a central stem provided with a screw, which passes through a hole and receives a nut applied from the under side. The nut has a flat-sided head, so that it may be readily operated with the fingers. A number of shallow grooves are cut around the outside of the piston to serve as water packing, so called.

The springs used in this indicator are of the duplex type, being made of two spiral coils of wire, with fittings, as shown in Fig. 153.

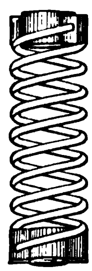


Fig. 153

The springs are adjusted under steam pressure, and are therefore correct only when used for steam engines. If required for water or other purposes, either special springs should be obtained, adjusted for the purpose for which they are required, or they should be tested at the time they are to be used and the actual scale of spring determined. It should be borne in mind that springs become impaired by long usage, and their scale changes in consequence. Therefore, for important work, the accuracy of the spring should always be tested by comparison on the spot with a reliable steam gauge, employing, as nearly as possible, the conditions under which the instrument was used. For steam work they may be tested by attaching to the main steam pipe, a half-inch pipe fitted with a globe valve, a tee for the attachment of the indicator, another tee for the attachment of the steam gauge, and finally a small drip valve. By keeping the drip valve slightly open, and regulating the globe valve, any desired pressure in the apparatus can be secured.

TABLE OF SPRINGS AND MAXIMUM PRESSURES FOR THE SAME.

Scale of Spring.	Maximum safe pressure to which a Spring can be subjected.	Scale of Spring.	Maximum safe pressure to which a Spring can be subjected.
8	10	40	100
10	15	48	120
12	20	50	125
16	24	60	150
20	40	64	160
24	48	80	200
30	70	100	250
32	75		

The paper drum E (as shown in Fig. 151) turns on a vertical steel shaft, secured at the lower end to the frame of the indicator. The drum is supported at the bottom by a carriage, which has a long vertical bearing on the shaft. It is guided at the top by the same shaft, which is prolonged for that purpose, the drum being closed at the top and provided with a central bearing. The drum is held in place by a close fit, and is easily removed by hand when desired. Stops are provided on the inside of the drum at the bottom, with openings in the carriage to correspond, so as to prevent the drum from slipping. These are so placed that the position of the drum may be changed so as to take diagrams in the reverse position of the pencil mechanism when so desired. The drum is made of thin brass tubing, so as to be extremely light. Steel clips are attached to the drum for holding the paper.

The drum carriage projects below the lower end of the drum, where it is provided with a groove for the reception of the driving cord. The drum spring, by which the backward movement of the drum is accomplished, consists of a flat spiral spring, placed in a cavity under the drum carriage encircling the bearing. It is attached at one end to the frame below and at the other end to the drum carriage. In its normal position, the drum carriage is kept against a stop, by means of the pull of the spring. By loosening a thumb screw which encircles the shaft and holds the drum carriage down to place, the carriage may be lifted so as to clear the stop, and the tension on the spring may then be adjusted. This is done by simply winding or unwinding as may be desired. A carrier pulley serves to operate the driving cord from any direction. A single pulley is mounted within a circular perpendicular plate, the center of which coincides with the center of the driving cord, and with the circumference of the pulley. The plate can be turned about its center so as to swing the pulley into any desired angular position, and thereby lead the cord off in any desired direction. The plate is held by a circular frame, which also serves as a clamp, and the pulley is fixed in position by the use of the same nut which secures the frame to the pulley arm. A ratchet is cut on the edge of the drum carriage, and a pawl is provided, attached to a frame, so as to engage in it whenever it is desired to stop the motion of the drum without unhooking the driving cord.

The indicator is attached by means of a coupling having but one thread. The indicator cock is provided with a stop, so as to turn only the 90 degrees needed for opening and closing. The pressure of the pencil on the paper drum is regulated by means of a screw, which passes through a projection on the slot plate, and strikes against a small slot provided for that purpose, which stands on the frame. This

screw is operated by a handle, which is of sufficient size to be readily worked by the fingers, and which also serves as a handle for turning the pencil mechanism back and forth, as is done in the act of taking diagrams.

The end of the pencil bar is shaped in the form of a thin tube for the reception of the pencil, lead or metallic marking point.

MANAGEMENT AND CARE OF THE INDICATOR.

The indicator should at all times be kept in good working order, and special attention should be directed to the condition of the cylinder and piston. These should particularly be kept free from any accumulation of dirt, for a little accumulation of dirt in the cylinder produces distortions in the diagrams and makes them almost valueless. The same result is produced by insufficient lubrication of the piston. The piston is easily removed by simply unscrewing the cover and lifting the whole from its place. After that is done introduce a cloth or cotton waste into the cylinder and clean it thoroughly. Then clean the outside of the piston and lubricate the cylinder and piston before replacing the piston. This should be done at frequent intervals, especially when the indicator is used continuously for a long test; in fact, the piston should be lubricated at least once for every half dozen diagrams taken. It will not do to use ordinary oil, but the best cylinder oil should be used. After cleaning the inside of the indicator and lubricating the piston and cylinder, the parts may afterward be lubricated without disconnecting them. Care must be taken to have perfect freedom in the pencil movement else the diagrams will be defective. The pencil bar, when not encumbered by a spring should, upon being raised to its highest position, drop like a dead weight to its lowest position with perfect freedom.

The surface of the roller and the sides of the slot should be kept clean, but should not be lubricated. The roller should be kept free and the pin oiled. The pivots at the joints in the pencil mechanism should be oiled from time to time. Examination should be made to see that a suitable amount of side play exists in these joints to insure their working with perfect freedom. All the working parts requiring lubrication except the inside of the cylinder should be lubricated with watch oil.

Due attention must be given to the paper drum as well as to the pencil mechanism. The paper drum is in constant motion when the indicator is set to work, and is therefore subjected to much wear. The drum carriage should be taken down from time to time and the central bearing cleaned. The top bearing of the drum and the carriage bearing

must also be cleaned; and all of these parts should be frequently lubricated.

The number of spring to be used in any particular case depends mainly on the boiler pressure; as a general rule, however, the lightest spring that can safely be employed is preferable, because the lighter the spring the higher will be the diagram, and hence the more accurate the measurements can be made. For the purpose of comparison it will be advantageous to employ the same spring, where admissible, in taking diagrams from various engines and under various conditions, as such comparisons can be made without making allowance for change in the tension of the spring. A No. 40 spring is a good standard in such cases for pressures below 80 or 90 pounds. For low-pressure cylinders in compound engines it is customary to employ springs below No. 20. For engines of high rotative speed it is well to employ springs of high tension to insure a smooth line in the diagram.

Springs are readily changed by removing the cover of the cylinder, loosening the screw beneath the piston, unscrewing the piston from the spring and the spring from the cover, and placing the spring desired. When the ball and socket end of the piston rod is introduced into the square hole in the center of the piston, care must be taken to have it set fairly in the hole before the screw is applied. If such care is not taken, the corners may catch and cause derangement.

To obtain well-defined diagrams, the pencil lead should be frequently sharpened. Very satisfactory results are obtained by the employment of a metallic marking point instead of a lead pencil, using chemically prepared paper on the drum. A piece of ordinary brass wire, suitably sharpened, will answer for the marking point. It requires less frequent sharpening than the lead pencil and the general character of the work done by it is much superior. It must be borne in mind, however, that lines made by a metallic point on chemically prepared paper will disappear in a short time. Therefore, in order to preserve the diagrams, the lines should be traced over with a pencil or a pen within a few days after they are made.

The tension on the drum spring should be adjusted so that the minimum force exerted by the spring may be sufficient to keep the cord taut. This, however, must be increased according to increased rotative speed, but under no circumstances should there be enough tension to make it difficult to pull the cord with the thumb and forefinger. The lightest tension consistent with good work is the most desirable. The tension on the drum spring may be adjusted by removing the paper drum, then loosening the thumb-screw which encircles the central shaft, lifting the drum carriage so as to clear the stop, and then winding the carriage in the desired direction.

HOW TO INDICATE A STEAM ENGINE.

The first thing to be done is to provide means for attaching the indicator to the cylinder of the engine; and, second, to provide means for giving motion to the paper drum.

To attach the indicator a hole should be drilled at each end of the cylinder, and tapped to fit a half-inch steam pipe, to which to connect the indicator cock. In horizontal engines the cylinder itself, instead of the head, should be drilled and tapped for the attachment of the indicator steam pipe. Care must be taken to drill the holes so that they can not be interfered with by the piston of the engine; and wherever the pipes are attached they should communicate freely with the steam in the cylinder throughout the entire length of the stroke of the piston; they should be short and free from any unnecessary bends. The pipes, bends, cocks or valves connecting the indicator with the cylinder should be so constructed and attached as to form a free and unobstructed passage of the steam in the engine cylinder to the piston of the indicator. The best results are not obtained by connecting the two ends of the cylinder with one indicator applied at the center, for the reason that it makes the pipes too long for accurate work, and necessitates the employment of extra or additional bends, all of which should be avoided, especially in indicating high-speed engines. In all such cases the indicator should be placed as near the cylinder as possible so as to reduce the length of the steam pipe to its minimum. If but one indicator is available it should be used first at one end, then at the other. Should there be a necessity for placing the indicator at the

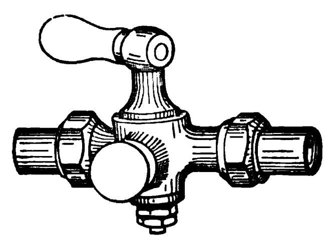


Fig. 154.
THREE-WAY COCK.

center of the cylinder, large pipes and easy bends will, to a great extent, avoid the errors due to long pipes. In such cases a three-way cock, such as shown in Fig. 154, should be employed.

HOW TO ATTACH THE INDICATOR.

In drilling and tapping holes in a cylinder care should be exercised that the chips do not enter it unless they can be removed. A simple

method for preventing chips going into the cylinder is to admit steam into it while the work of drilling and tapping the holes is being done, so that the chips will be blown out as fast as they are made.

Before attaching the indicator the cock should be opened and the pipes blown out and thoroughly cleared of all loose material that may have collected or lodged in them.

The motion to be given to the paper drum is one that coincides with the motion of the piston, except that it is on a reduced scale. This motion may be obtained in various ways, one of which is shown in Fig. 155.

REDUCING LEVER.

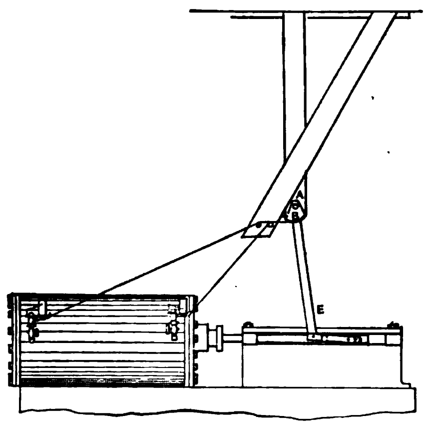


Fig. 155.

The reducing lever E consists of a strip of pine board three or four inches in width, about one and one-half times the stroke of the engine in length. It is hung by a screw or small bolt to a wooden frame attached overhead. At the lower end a connecting rod C D, is attached, about one-third as long as the stroke, one end to the lever and the other to a stud screwed into the cross head. The lever E should stand in a vertical position when the piston is at the middle of its stroke. The connecting rod at C, when the piston is in the middle of its stroke, should be as far below a horizontal line as it would be above it at either end of the stroke. The cords which operate the paper drum may be attached to a screw inserted in the lever E near the point of its suspension; but a better plan is to provide a segment A B, the center of which coincides with the point of suspension, and allows the cord to pass around the circular edge. The distance from edge to center should bear the same proportion to the length of the reducing lever that the desired length of the diagram bears to the length of the stroke of the engine. For an engine having a stroke of

48 inches, the lever should be 72 inches in length, and the connecting rod C D 16 inches in length; in which case, to obtain a diagram 4 inches long, the radius of the segment would be required to be 6 inches. It is, however, immaterial as to the actual length of the diagram, but 4 inches is usually satisfactory, but for high speeds it may be advantageously reduced to 3 inches. The cords should leave the segment in a line parallel with the axis of the cylinder. The pulleys over which they pass should incline from a vertical plane and point to the indicators wherever located. If the indicators and reducing levers can be placed so as to be in line with each other, the pulleys may be dispensed with, and the cords carried directly from the segment to the instruments, a longer arc being provided for that purpose. The carrier pulley on each indicator should be adjusted so as to point in the direction in which the cord is received.

In arrangements of this kind the reduced motion is not mathematically correct, because the leverage is not constant at all points of the stroke. Pantagraph motions have been devised for overcoming these defects. Two forms have been successfully used, which, if well made, well cared for, and properly handled, will reproduce the motion on the reduced scale with perfect accuracy.

LAZY TONGS.

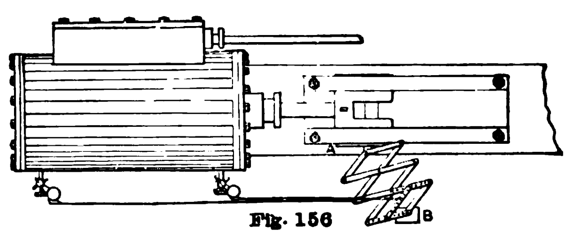


Fig. 156

Fig. 156 represents the manner of attaching the lazy tongs, when the indicators are applied to the side of the cylinder. It works in a horizontal plane. The pivot end is supported by a post B, erected in front of the guides, and the working end receives motion from an iron attached to the cross head. By adjusting the post to the proper height, and at a proper distance in front of the cross head, the cords may be carried from the cord pin E (Fig. 157) to the indicator without the use of carrier pulleys. As it is not always convenient to adjust the position and height of the post in this manner, a second post is required for the attachment of a pulley over which the cord passes. The posts can be erected on a board base, which can be fastened to the floor, and the posts rigidly held in place by wooden braces.

Fig. 157 shows a perspective view of the lazy tongs on a larger scale, detached from its working position on the engine. It consists of strips of wood put together at the pivot ends with hollow pins and large riveted heads The end B is attached to the stationary post by

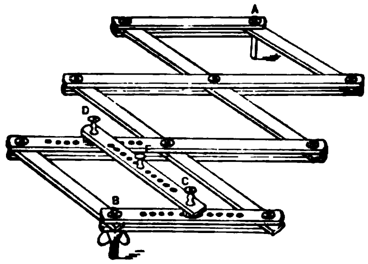


Fig. 157.

an iron stud, to which it is firmly held by means of a nut having flat sides, so as to be worked with the fingers. The end A drops into a hole in the iron attached to the cross head of the engine. The cord pin E is attached to a cross bar C D, which may be moved into different positions with relation to the center B, by changing the screws C and D which hold it in place. Holes are provided in the cross bars for various positions of the cord pin, corresponding to the various positions of the cross bar, this pin being always placed in line between the points A and B. The holes are one-half inch apart, and a change of one hole produces a change of one-forty-eighth in the length of the motion of the cord pin. In the hole nearest the center B the reduction of motion is three-forty-eighths, or one-sixteenth; in the second hole it is four-forty-eighths, or one-twelfth; in the third hole it is five-forty-eighths, or a fraction over one-tenth; in the fourth hole it is six-forty-eighths, or one-eighth; and so on.

The joints of the lazy tongs can be readily tightened when they become loose by upsetting the rivet heads. In using the instrument the joints should be kept well lubricated. A good plan is to keep the whole apparatus when not in use in a bath of oil

THE PANTAGRAPH.

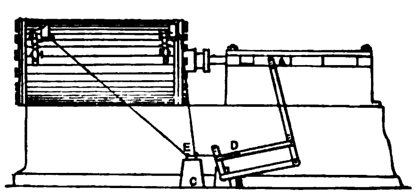


Fig. 158.

Fig. 158 shows a simple form of pantagraph, for use when the indicators are attached to the side of the cylinder. The working end A receives motion from the cross head, and the foot piece B is attached to the floor. The cord pin D is fixed in line between the pivot B, and the working end A, and the pulleys E, attached to the block G, guide the cords to the indicators.

The indicator rigging that gives the best results at high speeds is a plain, reducing lever, provided at the lower end with a slot that rides on a stud screwed into the cross head. The length of the lever should be two and one-half times the length of the stroke.

Whatever plan is adopted for an indicator rig, long stretches of cord should be avoided. If the motion must be carried a long distance it is desirable to use a fine annealed wire in place of the cord, except at the point where it passes over the pulley. Braided linen cord, a little in excess of one-sixteenth of an inch in diameter, is a suitable material for indicator work.

MANNER OF TAKING DIAGRAMS.

Stretch a blank card smoothly around the paper drum and fasten the ends with the spring clips. Attach the driving cord so that the motion of the drum will be central. Open the cock to admit steam to the indicator till the parts have become heated. Shut the steam off, bring the pencil or marking point in contact with the paper, adjust the top screw and trace a fine line upon the card. This is the atmospheric line. Open the cock, and after two or three revolutions, apply the pencil to the card and take the diagram. If it is desired to ascertain the condition of the valve adjustment the pencil need only be applied while the engine is making one revolution. But to determine the power it should be applied a longer time so as to obtain a number of diagrams superimposed on the same card. The fluctuations in the admission of steam, produced by governors which do not regulate closely, are so common that this course should always be pursued to obtain average results. After the diagram has been traced and the cock has been closed, the pencil should be applied lightly to the paper to see whether or not the position of the atmospheric line has been changed. If a new line has been traced it is evidence of error or derangement, and the operation should be repeated on a new card.

In adjusting the valves the boiler pressure should be observed, and the changes that are made before taking a diagram should be noted on the card for reference. If a series of diagrams are being taken for power, they should be numbered in order and the number of revolutions per minute noted on every card. If tests are to be made for power used by machines or tenants, a number of diagrams should be taken under each

condition and the results averaged. It is well, in such cases, to mark each card of a set by some letter, and on the first of the set specify the machines in operation at the time.

When taking diagrams care should be taken to see that the pressure of the pencil on the paper is just sufficient to make a legible mark, and no more; greater pressure only increases friction and produces inaccuracy in the diagram. Accumulation of water in the indicator is a hindrance to the taking of an accurate diagram; for that reason the indicator should be thoroughly heated and freed from any accumulation of water before taking a diagram. If much water is entrained in the steam the cylinder cocks should be kept slightly open while taking diagrams, otherwise they are likely to be distorted.

When taking diagrams from steam engines the height of the barometer, or pressure of the atmosphere, should always be carefully noted. This is necessary when the economy of the engines is to be determined, and it is desirable in all cases to know how much the exhaust pressure is above zero. Even at the sea level the pressure is constantly changing, and there are many engines working in places far above the sea level where the atmospheric pressure is always less, in some cases very much below 14.7 pounds per square inch, or 29.9 inches of mercury. Care should therefore be exercised in this respect, as many engineers are inclined to ignore this fact.

All steam gauges in ordinary use indicate pressures above the atmospheric pressure Vacuum gauges indicate the pressure below the atmospheric pressure. Neither kind, however, show the pressure above zero, or total pressure, and to arrive at this the pressure of the atmosphere must be added to the steam gauge pressure in the first case, or the amount of vacuum subtracted from the atmospheric pressure in the second.

THE ESSENTIAL FEATURES OF THE INDICATOR DIAGRAM.

The shape of the figure traced upon the indicator card depends upon the manner in which the steam pressure acts in the cylinder. If the steam be admitted at the beginning, and exhausted at the end of the stroke, and admission continued from one end of the stroke to the other, the shape of the diagram will be nearly rectangular. If the admission continues through only a part of the stroke, the diagram will assume a shape similar to that in Fig. 159.

Fig. 159 has been selected to illustrate the essential features in the indicator diagram, because it exhibits clearly all the operations effected by the pressure that commonly take place in a steam engine cylinder. This diagram shows that the admission of steam commences at A and ends at D; the cut off commences at C and becomes complete at D;

expansion occurs from D to E; the release or exhaust begins at E and continues to the point H; the compression of the exhaust steam commences at G and ends at the admission point A.

The line A B is called the admission line; B C the steam line; D E the expansion line; F G the exhaust or back pressure line—or in the case of condensing engines, the vacuum line; H A the compression line; and J I the atmospheric line. The curve which joins two adjacent lines, represents the action of the steam when one operation changes to another and can not properly be classed with either line.

The point of cut off, D, lies at the end of admission; the point of release, E, at the beginning of the exhaust; the point of compression, H, at the end of the exhaust. The proportion of the whole

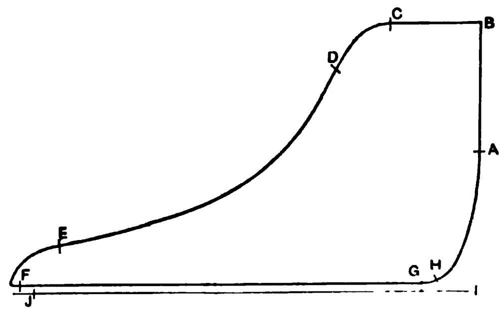


Fig. 159.

length of the diagram borne by the distance of the point D from the admission end, represents the proportion of the stroke completed at the point of cut off; so also in the case of the point of release, and in that of compression for the uncompleted portion of the stroke. The pressure at the points of cut off, release, and compression, are the heights of these various points above the atmospheric line measured on the scale of the spring.

There are three principal objects for which the indicator diagram is used:

First, to serve as a guide in setting the valves of an engine.

Second, to determine the indicated power developed by an engine.

Third, to determine, in connection with a feed-water test showing the amount of steam consumed, the economy with which an engine works.

A WELL-FORMED DIAGRAM.

Fig. 159 shows the general features of a well-formed diagram, the attainment of which should be the aim in setting the valves of an engine. The admission of steam is prompt, making the admission line perpendicular to the atmospheric line. The initial pressure is

fully maintained up to the point where the steam begins to be cut off. The somewhat early release secures a free exhaust and uniformly low back pressure, and the exhaust valve closes before the return stroke is completed, thus providing for compression. These are the first requirements to be met in producing an economical engine.

DERANGEMENT OF VALVE GEAR—HOW REVEALED.

Derangement in the valve gearing is revealed in the diagram by tardy admission or release; by low initial pressure or high back pressure; or by absence of compression; either one of which causes an increased consumption of steam for performing the same amount of work.

The angular position of the eccentric controls all the movements of the valves, but improper lengths of the eccentric rods, or improper proportions of lap and lead, or wrong position of the eccentric, will produce some of the faults mentioned.

REGULATING THE EXHAUST.

In regulating the exhaust of an engine, the utility of employing compression should not be overlooked. In the first place, it serves to overcome the momentum of the reciprocating parts of the engine, and to reduce the strain upon the connections caused by the sudden application of the pressure at admission. In the second place, compression is desirable on the ground of economy in the consumption of steam. It fills the wasteful clearance spaces of the cylinder with exhaust steam instead of filling them with live steam from the boiler. Compression produces a loss by the increased back pressure which it creates, but the loss is more than counterbalanced by the gain resulting from the reduction of clearance waste. But it is not advisable to compress above the boiler pressure.

PIPES AND PASSAGES FOR ADMISSION AND EXHAUST OF STEAM.

The valves being in proper adjustment the indicator diagram shows whether or not the pipe and passages for the admission and exhaust of steam are of sufficient size. In automatic cut-off engines the admission line should be parallel with the atmospheric line, and the initial pressure should not be more than three pounds less than the boiler pressure. The back pressure should not, in any engine, exceed one pound, where the exhaust proceeds directly to the atmosphere. Much can often be learned by applying the indicator to the steam and exhaust pipes, using the same mechanism for driving the paper drum as that used when the indicator is operated at the cylinder.

EXAMINATION OF VALVES OF ENGINES LONG IN USE.

Before making adjustments upon engines that have been long in use, it should be ascertained whether a valve, which should travel in a different place, has worn to a shoulder upon its seat. If changed under such circumstances, loss from leakage may follow, and be sufficient in amount to more than counterbalance the saving that might otherwise result.

USEFULNESS OF THE INDICATOR.

The indicator is useful in determining the amount of power developed by the engine. The diagram reveals the force of the steam at every point of the stroke; the power is computed from the average amount of this force, which is independent either of the adjustment of the valves, the form of the diagram or of any condition upon which economy depends. The diagram gives what is termed the *indicated* power of an engine, which is the power exerted by the steam. The indicated power consists of the net power delivered, and in addition, that consumed in propelling the engine itself.

The indicator is useful in determining in connection with a feed-water test, the number of pounds of steam consumed by an engine per indicated horse power per hour. This quantity forms a measure of the performance of an engine, and when compared with the performance of the best of its class, shows the economy with which the engine works. The amount of steam consumed is usually found by weighing the feed water before it is supplied to the boiler. This requires the erection of a weighing apparatus, which generally consists of two tanks and a platform scales. One tank is placed on platform scales elevated above the other tank, the latter tank being larger than the former. The water is drawn into the first tank and weighed; it is then discharged into the second tank and from this it is pumped into the boiler.

Feed-water tests are of no value when made by measuring the water fed to the boiler unless leakage of water from the boiler, if any exist, is allowed for. Strict attention should always be given to this important matter, and the rate of leakage should be determined by observing the fall of water in the gauge when no steam is being drawn from the boiler, a constant pressure being maintained.

A portion of the feed water consumption of an engine may be found by computation from the diagram without the aid of a feed-water test. The manner of making this computation will be found in the chapter on "The Buckeye Engine." If it were not for the losses produced by leakage and cylinder condensation, the whole amount of feed water consumed might be determined in this manner. Leakage

of steam often occurs and cylinder condensation is inevitable, yet the extent to which these losses act is not revealed by any marked effect produced upon the lines of the diagram. The measurement of the consumption of steam by diagram can not, therefore, be taken to show actual performance without allowance for these losses. But diagrams taken in connection with feed-water tests will reveal the extent of the losses produced by leakage and cylinder condensation. These losses are represented by that part of the feed water consumption which remains after deducting the steam computed from the diagram, or accounted for by the indicator as it is termed. The loss by condensation is nearly constant for different engines working under similar conditions, and an allowance may therefore be made for its amount. The other leakage, depending upon the wearing surface of valves, piston and cylinder, is variable in different cases. The fact of the presence of such leakages may be detected by a trial under boiler pressure with the engine at rest, the leakage being revealed by escape at the indicator cock or exhaust pipe. The amount of this leakage may be determined by computing that part of the loss not covered by condensation. In other words, in the case of leaking engines, when the indicator and feed-water tests show that there is more loss than is produced in good practice by condensation, the excess represents the probable amount of loss by leakage. A valuable use for the indicator is thus found in connection with feed-water tests. To make it available in practice, tables A, B and C are appended showing the percentages of loss that occur from cylinder condensation.

TABLE A.

PERCENTAGE OF LOSS BY CYLINDER CONDENSATION TAKEN AT CUT OFF IN SIMPLE ENGINES.

Percentage of Stroke Completed at Cut Off.	Percentage of Feed Water Consumption Accounted for by the Indicator Diagram.	Percentage of Feed Water Consumption Due to Cylinder Condensation.
5	58	42
10	66	34
15	71	29
20	74	26
30	78	22
40	82	18
50	86	14

The quantities in Table A apply to the type of unjacketed simple engines, in common use, having cylinders exceeding 20 inches in diameter.

TABLE B.

PERCENTAGE OF LOSS BY CYLINDER CONDENSATION TAKEN AT CUT OFF IN THE HIGH PRESSURE CYLINDER—IN COMPOUND ENGINES.

Percentage of Stroke Completed at Cut Off.	Percentage of Feed Water Consumption Accounted for by the Indicator Diagram.	Percentage of Feed Water Consumption Due to Cylinder Condensation.
10	74	26
15	76	24
20	78	22
30	82	18
40	85	15
50	88	12

The quantities in Table B apply to compound engines of the best class, having steam jacketed cylinders.

TABLE C.

PERCENTAGE OF LOSS BY CYLINDER CONDENSATION TAKEN AT CUT OFF IN THE HIGH PRESSURE CYLINDER—IN TRIPLE EXPANSION ENGINES.

Percentage of Stroke Completed at Cut Off.	Percentage of Feed Water Consumption Accounted for by the Indicator Diagram.	Percentage of Feed Water Consumption Due to Cylinder Condensation.
15	78	22
20	80	20
30	84	16
40	87	13
50	90	10

The quantities in Table C apply to triple expansion engines of the best class, having steam jacketed cylinders, supplied with dry but not superheated steam.

As an example of the use of the tables, Table A will be taken as an illustration of a test made on a 16 inch non-condensing engine of the automatic cut-off four-valve type. A feed-water test gave a consumption of 32.61 pounds per indicated horse power per hour. The cut off was 23.7 per cent., and the steam accounted for by the indicator was 69.2 per cent. Referring to Table A it appears that good practice calls for a considerable larger per cent. to be accounted for. At 23.7 per cent. cut off the amount is about 75 per cent. A leakage test of the valves and piston showed that the steam valves were tight, and that the piston was practically tight. The exhaust pipes, however, leaked badly, a considerable blow of steam issuing from the exhaust pipes when pressure was applied to them. Evidently leakage was the

cause of the low percentage of feed water accounted for by the diagram, its extent being represented by the difference between 69 per cent. and 75 per cent. The leaking valves were repaired, after which a feed-water test gave a consumption of 29.37 pounds per indicated horse power per hour. The cut off was 22.3 per cent. of the stroke, and the percentage of steam accounted for by the diagram was brought up to 74.5 per cent. or practically the figure given in the table.

COMBINING THE DIAGRAMS FROM COMPOUND ENGINES.

There are certain losses in simple or non-compound engines which compounding corrects in a great part, but in turn introduces other losses, which it is desirable to reduce to the least amount possible. These are the losses between the two cylinders, and consists of condensation in the passages, pipes and receiver—if one is used—friction in the parts and pipes, and expansion of steam taking place between the two cylinders without doing useful work. The extent of these losses can be shown by combining the diagrams from the two cylinders and drawing in the hyperbolic curve. This curve should just touch the expansion line of the high-pressure diagram at the point where the exhaust from that cylinder begins, and the space between the curve and both diagrams below this point and also the space between the two diagrams represent the loss between the two cylinders.

To correctly combine the two diagrams the clearance in each cylinder should be known and accounted for, as well as the piston displacement, and the relative length of the two diagrams when combined is the ratio of the volume of the two cylinders, that is, the piston displacement plus the clearance at one end. It is best to decide on the total length of the low-pressure diagram first, and a length that can easily be divided into one hundred parts will be found most convenient, as percentages of this length can be easily measured; 10 inches for a scale of tenths, or 12½ inches for a scale of eighths, are fair lengths.

Decide on the scale of pressure to which the two diagrams are to be plotted—usually it will be most convenient to take the scale of the original low-pressure diagram—draw in the atmospheric and vacuum lines and erect perpendiculars at the two extremes of the combination diagram, one of which is the clearance line. All measurements of distances should be made from the clearance line, and all measurements of pressure from the atmospheric line. Divide each original diagram into any desired equal parts, say ten, as that is a good number. Find the volume of piston displacement of the low-pressure cylinder, and add to it the volume of clearance; the total length of the diagram represents this total volume. Divide the clearance by the total volume, and the quotient will give the percentage this clearance bears to the

whole length. Set off this distance from the clearance line, and divide the remainder—representing the piston displacement—into the same number of equal parts that the original diagram is divided into. If the scale selected is the same as that of the original diagram, transfer the pressures directly with a pair of dividers from the lines on the diagram to the corresponding lines on the combination; draw in the connecting portions of the diagram, and the result will be an elongated diagram from the low-pressure cylinder, or as if it had been taken with the same spring as before, but with greatly enlarged paper drum.

Find the total volume of the high-pressure cylinder and divide by the total volume of the low-pressure cylinder, and the quotient will give the percentage of length of the diagram, which should be measured from the clearance line. Divide the clearance volume of the high-pressure cylinder by the total volume of the low-pressure cylinder, and measure off the percentage of length as before from the clearance line. Divide the remaining length of the high-pressure diagram—representing the piston displacement—into the same number of equal parts as the original diagram, and transfer the pressure from the lines on the original diagram to the corresponding lines on the combination and the new scale of pressures. If the original high-pressure diagram was taken with a No. 40 spring, and the combination diagram made to a scale of 10 pounds per inch, the new diagram will be four times as high as before, although it may be shorter.

Next draw the hyperbolic curve as shown in Fig. 162.

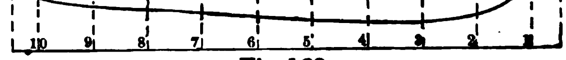


Fig. 160.

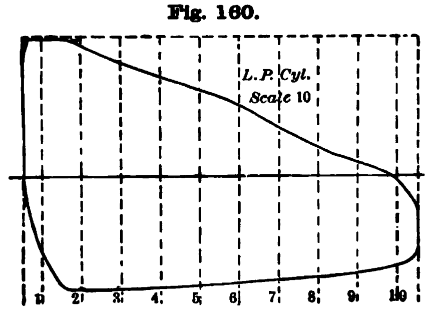


Fig. 161.

High-pressure cylinder, diameter 30 inches.
Low-pressure cylinder, diameter 50 inches.
Boiler pressure, 82 pounds.
Revolutions of engine per minute, 56½.
Vacuum, 12 pounds.

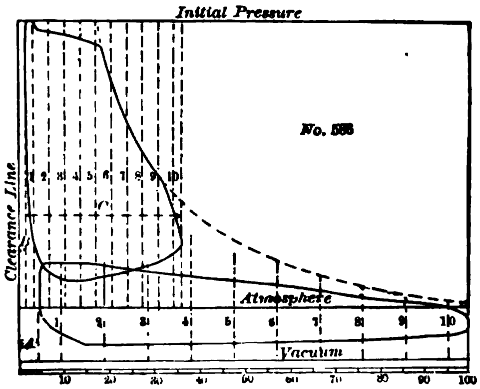


Fig. 162.

Fig. 162 is a combination of 160 and 161, taken from a tandem compound engine.

In order to make the explanation clearly understood, the diagrams in Figs. 160, 161 and 162 have the construction lines shown, and the necessary calculations are given below:

The combination diagram shown in Fig. 162, was drawn 12½ inches long over all, and by photographing reduced to its present length.

Diameter of high-pressure cylinder, 30 inches.
Diameter of low-pressure cylinder, 50 inches.
Stroke of piston, 72 inches.
Diameter of piston rod, both cylinders, 6¼ inches.

VOLUME OF CYLINDERS.

TO DETERMINE THE VOLUME OF HIGH AND LOW PRESSURE CYLINDERS.

RULE.—First, determine the area of the diameter of the cylinder; then determine the area of the diameter of the piston rod; then subtract the area of the piston rod from the area of the cylinder, and multiply the remainder by the length of the stroke of the piston, in inches, and the product will give the volume of piston displacement for that stroke. The cards being taken from the piston rod end.

Second, add the volume of clearance to the volume of piston displacement, and the sum will give the total volume of the cylinder.

VOLUME OF HIGH-PRESSURE CYLINDER.

$$\overline{(30\times 30\times .7854)-(6.25\times 6.25\times .7854)}\times 72=48684.9825 \text{ cubic inches.}$$

Volume of piston displacement of high-pressure cylinder.

Volume of piston displacement of high-pressure cylinder.	48684.9825
Clearance volume of high-pressure cylinder.	2545
Total volume of high-pressure cylinder.	51229.9825

VOLUME OF LOW-PRESSURE CYLINDER.

$$\overline{(50\times 50\times .7854)-(6.25\times 6.25\times .7854)}\times 72=139163.0625 \text{ cubic inches.}$$

Volume of piston displacement of low-pressure cylinder.

Volume of piston displacement of low-pressure cylinder.	139163.0625
Clearance volume of low-pressure cylinder.	7673
Total volume of low-pressure cylinder.	146836.0625

COMPUTING THE HORSE POWER OF AN ENGINE FROM THE INDICATOR DIAGRAM.

The work done by the steam in the cylinder of an engine is measured by the product of the force exerted on the piston, into the distance through which the piston moves, and is usually expressed by the term foot pounds. If, for example, a force of 33 pounds per square inch on a piston having an area of 100 square inches is employed to drive the piston 100 times over a stroke of 4 feet, the work done by the steam amounts to $33\times 100\times 100\times 4=1{,}320{,}000$ foot pounds. The amount of horse power which the steam develops is the foot pounds of work done in a minute divided by 33,000. In the example given, the horse power developed when 100 strokes are made per minute is 1,320,000 divided by 33,000. Thus:

$$1{,}320{,}000\div 33000=40 \text{ horse power.}$$

The force exerted by the steam upon the piston is given by the indicator diagram, but as it varies in amount in different points of the stroke, it is necessary to determine the equivalent force which, acting constantly, would produce the same result. This is done by computing from the diagram what is termed the

MEAN EFFECTIVE PRESSURE.

The product of the mean effective pressure, expressed in pounds per square inch; the area of the diameter of the cylinder, expressed in square inches; the length of the stroke, expressed in feet; and the number of strokes per minute, which is twice the number of revolutions of the crank per minute, gives the number of foot pounds of work performed per minute. This result, divided by 33,000 gives the amount of horse power developed.

To compute from the diagram the mean effective pressure, two lines are drawn perpendicular to the atmospheric line, one at each end

of the diagram, and the intermediate space divided into 10 equal parts, with a perpendicular line at each point of division, as shown in Fig. 163.

A ready method of performing the division is to lay upon the diagram a scale of 10 equal parts, the total length of which may be a small amount in excess of the length of the diagram. It is so placed in a diagonal position that the extreme points on the scale lie upon the two outside perpendicular lines. The desired points may then be dotted with a sharp pencil opposite the intermediate divisions on the scale, as shown in Fig. 163. The points where the lines of division cross the diagram should be dotted; and in locating these points they

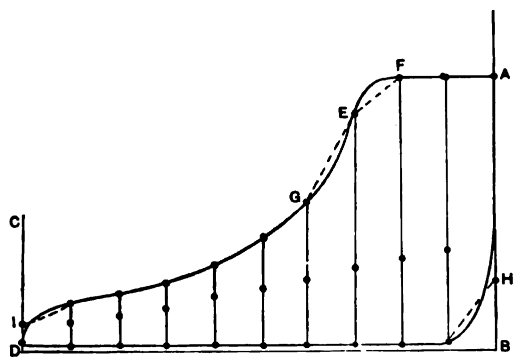


Fig. 163.

should be so placed that the area of the figure enclosed by a straight line joining them is exactly equal to the area enclosed by the curved line of the diagram. The proper location can be readily determined by the eye.

Fig. 163 shows the extreme perpendicular lines A B and C D, the intermediate lines of division, the points of intersection, and those points which require special location, as for example, the one at E, which is so placed that the area enclosed by the dotted line E F and E G is equal to that enclosed by the diagram from F to G.

The determination of the mean effective pressure now consists in finding the average length of the various perpendicular lines included between the points of intersection measured on the scale of the spring. This may be done by measuring each line with the scale and averaging the results. A better and quicker method is to employ a strip of paper and mark one after another of the various distances on its edge, making a mechanical addition and requiring only a final measurement with the scale of the spring and then dividing it by the number of spaces in the diagram. The proper course to pursue is to lay the edge of the paper on the first line and mark the distance from A to H,

shown in the diagram (Fig. 163), starting from the end of the measuring paper. Transfer the paper to the last line, and add to the first measurement the distance from I to D. Mark off from the end of the paper one-half the sum of these two distances, and from the middle point continue the addition of the intermediate nine divisions. When all have been marked, measure with the scale of the spring, from the end of the paper to the last addition, and divide the result by 10. This gives the mean effective pressure. It is essential that *one-half* the sum of the first and last distances on the diagram be taken, and this, together with the sum of distances of intermediate lines, be divided by 10. It is an error to divide by 11, as is sometimes done.

EXPANSION LINE IN NON-CONDENSING ENGINES CARRYING A LIGHT LOAD.

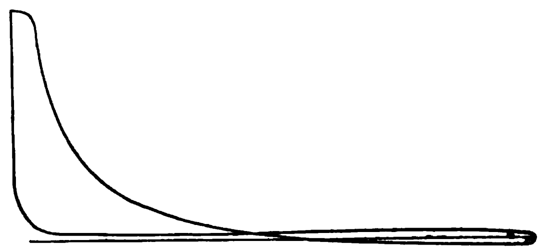


Fig. 164.

In non-condensing engines carrying a light load the expansion line often extends below the atmospheric line, as shown in Fig. 164. In computing the mean effective pressure in such cases, the distances below the atmospheric line must be deducted from those above it. The sum of the first and last distances here becomes the difference between the first and last.

PLANIMETERS.

When the mean effective pressure on a large number of diagrams is desired, much time and labor may be saved by the employment of a planimeter, an instrument designed to measure the areas of irregular figures. It is operated by moving a tracer, with which it is fitted, over the line of the diagram, and it records the area upon a graduated wheel. One of these devices is the Coffin Averaging Instrument, shown in Fig. 165.

In using this planimeter the grooved metal plate I is first connected to the board upon which the apparatus is mounted, in the position shown in the engraving, being held in place by a thumb-screw applied from the back side. The indicator card is then placed under

the clamps C and K, which may be sprung away from the board a sufficient amount to allow the card to be introduced, and the card is moved toward the left into such a position that the atmospheric line is near to and parallel with the lower edge of the stationary clamp C, while the extreme left-hand end of the diagram is even with the perpendicular edge of the clamp. The movable clamp, K, which is fastened at the bottom to a sliding plate, is then moved toward the left until the vertical beveled edge just touches the extreme right-hand end of the diagram. The diagram shown in the engraving represents the proper

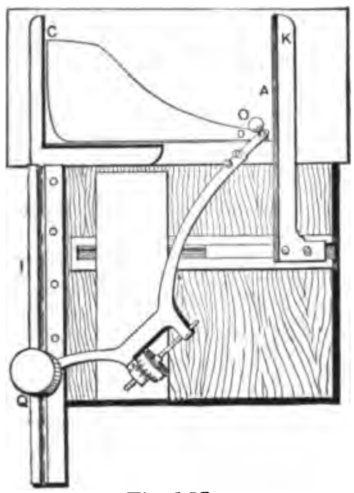


Fig. 165.

location which should exist when these preliminary adjustments have been completed. The slide at the bottom of clamp K fits closely, so that the application of a slight pressure is required to displace it.

The beam of the instrument is next placed on the board with the pin at the lower end resting in the groove I, and the weight Q applied to the top of the pin so as to keep it securely in place. The tracer O is moved to the right-hand end of the diagram and set at the point D on the line of the diagram, where the clamp K and the diagram touch each other. Here a slight indentation is made in the paper by pressing the finger on the top of the tracer, and this serves as a starting point. The graduated wheel is next turned so as to bring its zero mark to the zero mark on the vernier. The instrument is now ready for operation. The tracer O is carefully moved over the line of the diagram, in the direction of motion of the hands of a watch, and continued until a complete circuit is made and the tracer finally reaches the starting point D. Keeping the eye on the wheel, the tracer is now

moved upward by sliding it along the edge of the clamp K until the reading on the wheel returns to zero. Another light indentation is made in the paper to mark the new position which the tracer occupies. This point is represented at A in the engraving. The instrument is now moved away, the clamp pushed back, and the distance between the two points D and A is measured, by employing a scale corresponding to the number of the spring used in the indicator. The distance thus found is the mean effective pressure, expressed in pounds per square inch of piston.

This planimeter determines the desired result without computation, but it may be used also for determining the area enclosed by the diagram. This area is given by the reading on the graduated wheel, when the circuit of the diagram has been made and the tracer reaches the starting point D. The wheel has fifteen main divisions, each of which represents one square inch of area. Each division has five subdivisions, each subdivision representing one-fifth, or two-tenths of a square inch of area. The vernier scale enables the subdivisions to be used to fifteenths, each of these fifteenths, therefore, representing two-one-hundredths of a square inch.. Having obtained the area in this manner, the mean effective pressure may be computed by dividing the number of the spring representing the pressure per inch in height by the length of the diagram, in inches, and multiplying the quotient by the area, in square inches.

In first placing the indicator card under the clamps, care must be observed that the ends of the diagram set a little way from the edge of the clamp, so as to allow for one-half of the diameter of the tracer, and to bring the center of the tracer over the center of the line of the diagram.

Another devise for measuring diagrams is the Amsler Polar Planimeter, as shown in Fig. 166.

This instrument does not give the mean effective pressure directly, but it determines the area of the diagram, and from this area the mean effective pressure is computed in the manner above pointed out.

The polar planimeter consists of two arms A and B, and the measuring wheel C. To operate it a piece of smooth, hard paper is laid on the table, and the instrument placed upon it, with the needle point A pressed into the board. This point serves as a center about which the apparatus is turned. The indicator card is laid under the tracer B and held either by tacks, which fasten it to the table, or what is quite sufficient, by the pressure of the fingers. The tracing point is set on the line of the diagram—say, near the middle of the steam line—and a slight indentation made in the paper to serve as a starting point. The graduated wheel is set at the zero mark. The tracer is then moved

over the line of the diagram in the direction of motion of the hands of a watch, finally making a complete circuit returning to the starting mark. The number of divisions and fractions of a division, shown on the wheel at the point opposite the stationary zero mark, indicates the area of the diagram traced. The wheel has ten main graduations, each of which represents one square inch of area. Each main division is subdivided ten times, and each subdivision represents one-tenth of a square inch of area. A stationary vernier scale is placed beside the graduated edge of the wheel, and serves to indicate the smaller fractions, viz., hundredths. To read the vernier, the eye is run along the

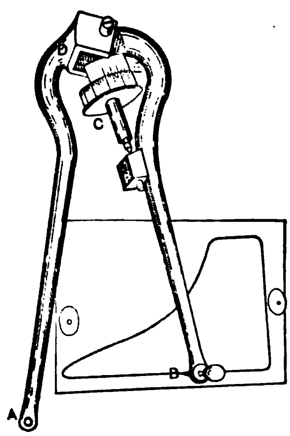


Fig. 166.

stationary scale until a line of division is found which is just opposite a division on the wheel. The number of the division on the vernier, reckoned from the zero mark, is the number of hundredths sought. If, for example, the reading of the area is two main divisions on the wheel, and four of the subdivisions and the line of coincidence on the vernier is No. 7 reckoned from zero, the area sought is 2.47 square inches. To reduce this to mean effective pressure, two perpendicular lines are drawn, one through each terminal point of the diagram, and the length of the diagram is measured by measuring the distance between these two perpendiculars. Suppose this distance is 3.78 inches, and suppose the number of the spring employed is 40. Then the mean effective pressure is found by dividing 40 by 3.78 and multiplying the quotient by 2.47, the product will be 26.13 pounds per square inch, the mean effective pressure.

In working the polar planimeter care must be taken to place the diagram so that the two arms are not brought too near each other at one end of the course, nor yet carried too far apart at the other end.

At a point midway between the two extremes the two arms should lie about parallel to each other.

As an example of the manner of computing the horse power of an engine, suppose an engine has a cylinder 15 inches in diameter, a piston rod $2\frac{1}{2}$ inches in diameter, a stroke of $2\frac{1}{2}$ feet, and engine running at a speed of 135 revolutions per minute. Suppose the indicator diagrams show a mean effective pressure of 36 pounds per square inch, this being the average of the indications at the two ends. The area of the cylinder to be used is the net area obtained by deducting one-half the area of the piston rod.

The area of a cylinder 15 inches in diameter is

$$15 \times 15 \times .7854 = 176.7150 \text{ square inches}$$

The area of the piston rod, $2\frac{1}{2}$ inches in diameter, is

$$2.5 \times 2.5 \times .7854 = 4.9+ \text{ square inches.}$$

One-half the area of the piston is

$$2)\underline{4.9}$$
$$2.45 \text{ square inches.}$$

The net area of the cylinder is

$$\begin{array}{rl} 176.7150 & \text{Area of cylinder.} \\ 2.45 & \text{Half the area of the rod.} \\ \hline 174.2650 & \text{Net area of cylinder.} \end{array}$$

The speed of the piston in feet per minute is

$$135 \times 2.5 \times 2 = 675 \text{ feet.}$$

The horse power developed is

$$\frac{174.265 \times 36 \times 675}{33000} = 128.3+ \text{ Horse power.}$$

RULE.—Multiply the net area of the diameter of the cylinder by the mean effective pressure per square inch, and multiply the product by the number of feet the piston traveled per minute, and divide the last product by 33,000, *and the quotient will give the horse power of the engine, as shown in the preceding and in the following examples.*

$$\frac{\text{Net area of cylinder in square inches} \times \text{mean effective pressure} \times \text{piston speed in feet per minute}}{33000} = \text{Horse power.}$$

When an engine has more than one cylinder, the power developed in each cylinder is computed according to the rule given, and the results added together.

CHAPTER XVI.

UNITED STATES MARINE BOILER INSPECTION AND ENGINEERS' LICENSE LAWS.

In entering upon this subject the student is warned against the practice of committing to memory so-called examination questions, and also in his wasting time in learning how to answer them; because the time spent in that direction neither tends to aid him in passing a successful examination before the inspection officers, nor to make him a safe or a practical engineer.

The examination of engineers under the steam boiler inspection laws of the United States, or of any of the States which have enacted laws upon that subject, are not based upon any fixed set of questions, nor upon any particular part of the profession of steam engineering; but they reach into and cover the whole field, and especially that relating to the management of steam boilers with safety to human life. This, then, is the sole object; it is the entire object in the passage of all laws relating to the construction, inspection and management of steam boilers, and to the examination and licensing of engineers.

The United States Government has set the example on this continent in the enactment of laws requiring the inspection of steam boilers and appurtenances therewith connected, and the examination and licensing of engineers.

Since then a number of States have followed the example of the Federal Government in providing, by law, for the examination and licensing of engineers, and some of the States have also included the inspection of steam boilers in their statutory enactments. It is, therefore, safe to predict that ere long every State in the Union will have similar laws placed upon its statute books.

As such laws are enacted for the sole purpose of protecting life and property, they necessarily exclude incompetent persons from having the care or management of steam boilers and steam engines. It is, therefore, of the utmost importance that all persons who desire to follow the profession of steam engineering should fit themselves for successful examination by the inspection officers. Aside from the requirements of the law, it is the duty of every engineer to acquire the knowledge and experience so indispensable in the prevention of steam boiler

accidents. Ignorance on the part of the boiler maker who constructs the boiler, or on the part of the engineer under whose care and management it is placed, is no excuse for the destruction of human life, even in the absence of any law excluding ignorant or incompetent persons from having the care or management of steam plants. Therefore, to enable engineers, and those who desire to become engineers, to take charge of and manage steam boilers and steam engines without hazard to human life, and to pass a successful examination before inspection officers, this chapter is devoted to the practical application of the rules and formulæ contained in the chapters which precede it.

As those laws will be taken as the standard by the various States in the enactment of inspection and license laws, and by all inspectors appointed and operating under them, the student will have the assurance, if he acquires the information contained in this work, that he will have no difficulty in passing a successful examination if he is a practical engineer; and in addition thereto he will have the satisfaction of knowing how to manage steam plants without endangering life or property.

In order to give full and complete information in regard to the requirements of the United States laws upon this subject, every section of the statutes, and every rule of the United States Board of Supervising Inspectors of Steam Vessels, requiring any mathematical calculations to be made by the United States inspectors of boilers, by candidates for the office of inspector, by marine boiler makers, and by engineers in their examination for engineers' license, will be separately and fully treated under this head.

QUALIFICATIONS OF INSPECTORS OF BOILERS.

Section 4415 of the Revised Statutes of the United States provides that an inspector of boilers shall be "a person of good character and suitable qualifications and attainments to perform the services required of inspectors of boilers, who, from his knowledge and experience of the duties of an engineer employed in navigating vessels by steam, and also of the construction and use of boilers, and machinery, and appurtenances therewith connected, is able to form a reliable opinion of the strength, form, workmanship, and suitableness of boilers and machinery to be employed without hazard to life, from imperfections in the material, workmanship, or arrangements of any part of such apparatus for steaming."

It is not only important that inspectors should thoroughly understand all the rules and regulations governing the construction and inspection of steam boilers to enable them to perform the duties pre-

scribed by law, but under department regulations, no person can be appointed inspector without first passing a most rigid examination, and no person can pass the required examination without a thorough knowledge of the requirements of the law as laid down and explained in this chapter.

QUALIFICATIONS OF ENGINEERS.

Section 4441 of the Revised Statutes of the United States provides that, "whenever any person applies for authority to perform the duties of engineer of any steam vessel, the inspectors shall examine the applicant as to his knowledge of steam machinery, and his experience as an engineer, and also the proofs which he produces in support of his claim; and if, upon full consideration, they are satisfied that his character, habits of life, knowledge, and experience in the duties of an engineer are all such as to authorize the belief that he is a suitable and safe person to be intrusted with the powers and duties of such a station, they shall grant him a license, authorizing him to be employed in such duties for the term of one year, in which they shall assign him to the appropriate class of engineers; but such license shall be suspended or revoked upon satisfactory proof of negligence, unskillfulness, intemperance, or the willful violation of any provision of this title."

It will be observed that the knowledge required on the part of inspectors of boilers and on the part of engineers covers completely the entire field of steam engineering so far as it relates to the safety of human life. In order, then, that an inspector shall have the qualifications prescribed by law he must be "*able to form a reliable opinion of the strength, form, workmanship and suitableness of boilers and machinery to be employed without hazard to life from imperfections in the material, workmanship, or arrangements of any part of such apparatus for steaming.*" It will also be noticed that the knowledge and experience required on the part of engineers are so interwoven with those required by inspectors, that the qualifications of persons desiring to become chief engineers of steam vessels require a knowledge and experience little short, if anything, of those required by inspectors. It is, therefore, impossible to separate the two, so far as the requirements of the United States laws are concerned, and the decisions and rules of the department having charge of the steam vessel inspection service require that no person shall be appointed inspector of boilers who has not held a license as chief engineer for a period of not less than three years, in addition to the qualifications prescribed by law. Therefore, the student of steam engineering, the inspector of boilers and the engineer are equally interested in the instructions given in this chapter, as neither can perform the duties

prescribed by law without the knowledge here given, nor perform the duties of an engineer without hazard to life or property. In order, then, that each may become familiar with the law, as well as to enable them to obtain the information each is required to possess, each section of the statutes, and each rule of the Board of Supervising Inspectors upon which any calculation is made, will be cited.

STRENGTH OF STEAM BOILERS.

[Section 4415 of the Revised Statutes of the United States.]

BURSTING PRESSURE OF SINGLE-RIVETED BOILERS.

RULE.—For boilers with single-riveted longitudinal seams in the cylindrical shell, multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength, in pounds per square inch; then divide the product by one-half of the diameter of the boiler, in inches, and multiply the quotient by the constant .56, the product will give the bursting pressure in pounds per square inch.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.
Let 60,000 pounds per square inch equal tensile strength of weakest plate.
Let 40 inches equal diameter of the boiler.
Let .56 equal a constant for single-riveted longitudinal seams.

Then we have:

$$\left(\frac{.25\times60000}{40\div2}\right)\times.56=420\text{ lbs.}$$ Bursting pressure per square inch.

Performing the operation in the ordinary way, we have:

		.25		Thickness of material in weakest plate.
		60000		Tensile strength of material per square inch.
Dividing by one-half of the boiler's diameter.	$40\div2=$ 20)	15000.00	(750	
		140	.56	A constant.
		100	4500	
		100	3750	
		0	420.00 lbs.	Bursting pressure per square inch.

BURSTING PRESSURE OF SINGLE-RIVETED BOILERS—SIMPLE RULE.

Another rule is as follows, and obviates the necessity of dividing the diameter of the boiler:

RULE.—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength, in pounds per square inch; then divide the product by the diameter (instead of half diameter as in the previous case) of the boiler, in inches, and multiply the quotient by 1.12, *the product will give the bursting pressure in pounds per square inch.*

Taking the same boiler employed in the previous example, we have:

```
                                   .25
                                  60000
                                 --------
Dividing by diameter of boiler. 40 ) 15000.00 ( 3 75
                                 120        1.12
                                 ---        ----
                                  300        7 50
                                  280       37 5
                                  ---      375
                                   200     ------
                                   200    420.00 lbs.  Bursting pressure per square inch.
                                   ---
```

BURSTING PRESSURE OF DOUBLE-RIVETED BOILERS.

What amount of pressure per square inch is required to burst a boiler with longitudinal seams double riveted?

RULE—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength, in pounds per square inch, and divide the product by one-half of the diameter of the boiler, in inches, and multiply the quotient by the constant .70; *the product will give the bursting pressure in pounds per square inch.*

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of weakest plate.

Let 48 inches equal diameter of the boiler.

Let .70 equal a constant for double-riveted longitudinal seams.

Then we have:

```
                                           .25
                                          60000
                                         --------
Dividing by one-half of  48÷2= 24) 15000.00 (625
the diameter of the                144         .70
boiler.                            ---        ------
                                    60      437.50 lbs.  Bursting pressure per square inch.
                                    48
                                   ---
                                   120
                                   120
                                   ---
```

Putting the operation in condensed form, we have:

$$\left(\frac{.25\times 60000}{48\div 2}\right)\times .70=437.50 \text{ lbs.}$$ Bursting pressure per square inch.

BURSTING PRESSURE OF DOUBLE-RIVETED BOILERS—SIMPLE RULE.

To obviate the necessity of dividing the diameter of the boiler, we perform the operation according to the following rule:

RULE.—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength, in pounds per square inch, and divide the product by the diameter of the boiler, in inches; then multiply the quotient by the constant 1.4; *the product will give the bursting pressure, per square inch, for boilers with double-riveted longitudinal seams.*

Taking the same boiler employed in the previous example, we have:

$$\left(\frac{.25\times 60000}{48}\right)\times 1.4 = 437.50 \text{ lbs. Bursting pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

		.25	Thickness of material.	
		60000	Tensile strength of material.	
Dividing by diameter of boiler.	48)	15000.00	(312.5	
		144	1.4	A constant.
		60	125 00	
		48	312 5	
		120	437.50 lbs.	Bursting pressure per square inch.
		96		
		240		
		240		

SAFE-WORKING PRESSURE OF MARINE BOILERS.

[Section 4433 of the Revised Statutes of the United States.]

This section of the Revised Statutes relates to boilers made of plates. It therefore does not include pipe or sectional boilers. Its provisions are as follows:

"SEC. 4433. The working steam pressure allowable on boilers constructed of plates inspected as required by this title, when single riveted, shall not produce a strain to exceed one-sixth of the tensile strength of the iron or steel plates of which such boilers are constructed; but where the longitudinal laps of the cylindrical parts of such boilers are double riveted, and the rivet holes for such boilers have been fairly drilled instead of punched, an addition of twenty per centum to the

working pressure provided for single riveting may be allowed: *Provided*, that all other parts of such boilers shall correspond in strength to the additional allowances so made; and no split caulking shall in any case be permitted."

The portion of this section relating to the drilling of rivet holes means that *all* rivet holes for boilers requiring the addition of twenty per cent. provided by law must be drilled. To emphasize this interpretation of the law, the Board of Supervising Inspectors of Steam Vessels has provided, by rule, that the additional twenty per cent. in the working steam pressure shall be allowed only where the longitudinal laps of the cylindrical part of the boilers are double riveted, and "*all* the rivet holes have been fairly drilled instead of punched."

It will therefore be observed that a boiler, in order to be allowed the additional twenty per cent., must have *all* of its rivet holes drilled as well as to have its longitudinal laps double riveted. Therefore, if a boiler has its longitudinal seams double riveted, and the holes in such seams have been drilled, it will not be allowed the additional twenty per cent. if the other holes have been punched instead of drilled. The student will therefore bear this in mind, as he proceeds with his studies of the requirements of the United States laws.

SAFE-WORKING PRESSURE OF SINGLE-RIVETED BOILERS.

What is the safe-working pressure for cylindrical boilers, or boilers having cylindrical shells, where the longitudinal seams are single riveted?

RULE.—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength, in pounds per square inch, and divide the product by the radius, or one-half of the diameter of the boiler, in inches, and then divide the quotient by 6 (which is called the factor of safety), and the last quotient will give the pressure, per square inch, allowable.

Example.—Let 25 one hundredths of an inch equal thickness of material.

Let 60,000 pounds per square inch equal tensile strength of material.

Let 40 inches equal diameter of the boiler.

Let 6 equal a factor of safety.

Then we have:

$$\frac{(.25 \times 60000) \div (40 \div 2)}{6} = 125 \text{ lbs.} \quad \text{Safe-working pressure per square inch.}$$

Putting the operation in the ordinary form, we have:

```
                                    .25       Thickness of material.
                                  60000       Tensile strength of material.
Dividing by one-half of the
diameter of the boiler.  40÷2= 20 ) 15000.00 ( 750  The quotient.
                                  140
                                  ---
                                   100
                                   100
                                   ---
                                     0
```

Then dividing the quotient by 6, the factor of safety, we have:

```
6)750
  125 lbs. Safe-working pressure per square inch
```

The law provides that no greater strain than one-sixth of the tensile strength of the material will be allowed. The examples given hardly make it clear that the pressure given in the answer produces no greater strain than that limited by law. This point will therefore require explanation.

The meaning of the law is that but one-sixth of the actual strength of the material will be allowed on the material. Material, then, having a tensile strength of 60,000 pounds per square inch of section, would, if it were one-fourth of an inch in thickness, and one inch in width, have an actual strength of 15,000 pounds, or one-fourth of 60,000 pounds; and such material would be allowed a strain of 2500 pounds, one-sixth of its actual strength. In other words, the law means one-sixth of the actual strength of the material in the plates—the strength of the seams is not taken into consideration except so far as limiting the pressure is concerned; but the law means the solid part of the sheets, and not the seams, when it refers to one-sixth of the tensile strength of the plates.

In order to make this plain to the student, a mathematical demonstration will be made.

ACTUAL STRENGTH OF BOILER PLATE.

RULE.—Multiply the tensile strength of the material, per square inch, by the thickness of the plates, in decimals of an inch, and the product will give the actual strength of the material one inch in width.

Example.—Let 65,842 pounds equal tensile strength per square inch of the material.

Let 25 one hundredths of an inch equal thickness of the plates.

Then we have:

```
  65842
    .25
 ------
 329210
131684
-------
16460.50 lbs.
```

Strength of material one inch in width and 25 one hundredths of an inch in thickness.

Taking the material employed in the boiler in the example preceding the last, in order to prove the correctness of the rule given for such cases:

Example.—Let 25 one hundredths of an inch equal thickness of material.

Let 60,000 pounds per square inch equal tensile strength of material.

Then we have:

```
     .25
   60000
--------
15000.00 lbs.
```

Tensile strength of material one inch in width and 25 one hundredths of an inch in thickness.

This material, then, would be allowed a strain of one-sixth of 15,000 pounds; and one-sixth of 15,000 pounds is

```
6) 15000
   -----
    2500 lbs.
```

Now, if we divide one-sixth of the strength of the material one inch in width and 25 one hundredths of an inch in thickness by one-half of the diameter of the boiler in inches, we have the pressure per square inch that will produce a strain on the plates equal to one-sixth of the tensile strength of such plates.

Thus: Taking a boiler 40 inches in diameter, we have:

$40 \div 2 = 20$ inches. One-half of the diameter of the boiler.

```
20) 2500 (125 lbs.
    20
    --
     50
     40
    ---
     100
     100
    ---
```

Pressure per square inch on a boiler 40 inches in diameter that will produce a strain on the material having a tensile strength of 15,000 pounds equal to one-sixth of that strength.

STRAIN PRODUCED ON THE PLATES BY THE PRESSURE PER SQUARE INCH IN THE BOILERS.

RULE.—Multiply the given pressure, in pounds per square inch, by one-half of the diameter of the boiler, in inches, and the product will give the exact strain on each inch in the entire length of the boiler.

Example.—Let 125 pounds per square inch equal given pressure.
Let 40 inches equal diameter of the boiler.

Then we have:

	125 lbs.	Given pressure per square inch.
$40 \div 2 =$	20 inches.	Half diameter of boiler.
	2500 lbs.	Strain produced on each inch in the length of the boiler.

To allow such a strain, the material would have to be six times stronger per square inch than 2500 pounds.

Hence: $2500 \times 6 = 15000$ lbs. Required tensile strength of material one inch in width and 25 one hundredths of an inch in thickness.

THICKNESS OF MATERIAL REQUIRED.

How thick would the material have to be with a tensile strength of 60,000 pounds per square inch, to have a tensile strength of 15,000 pounds in a width of one inch?

RULE.—Divide the tensile strength of the material one inch in width by the tensile strength of the material per square inch, and the quotient will give the thickness required in decimals of an inch.

Example.—Let 15,000 pounds equal tensile strength of material one inch in width.
Let 60,000 pounds equal tensile strength of the same material one square inch.

Then we have: $\frac{15000}{60000} = .25$ Decimals of an inch. Thickness of material required.

Performing the operation in the ordinary way, we have:

$$
\begin{array}{r}
60000)\ 15000.0\ (0.25 \\
12000\ 0 \\
\hline
3000\ 00 \\
3000\ 00 \\
\hline
\end{array}
$$

Decimals of an inch. Thickness of material required.

To sum the matter up briefly, the student is informed that the safe-working pressure allowable, under the law, for single-riveted boilers, is simply a pressure per square inch that will produce a strain on the material in the boiler not exceeding a strain equal to one-sixth of the strength of that material. This must not, however, be confounded

with the strength of the material per square inch—that is, if it was one inch thick—because the law simply means one-sixth of the strength of the material just as it is in the boiler, no matter what its thickness may be; and when adding twenty per cent. for double riveting, the law simply allows an addition of twenty per cent. on that allowed for single-riveted boilers.

SAFE-WORKING PRESSURE OF DOUBLE-RIVETED BOILERS—HOLES DRILLED.

[Section 4433 of the Revised Statutes of the United States.]

What is the safe-working pressure for a boiler with double-riveted longitudinal seams and all the rivet holes drilled?

RULE.—Multiply the thickness of material in the weakest plate in the boiler, in decimals of an inch, by the tensile strength, in pounds per square inch, and divide the product by one-half the diameter of the boiler, in inches; then divide the quotient by 6, and then add twenty per cent. to the last quotient; the sum will give the pressure, per square inch, allowable under the United States rule.

Example.—Let 25 one hundredths of an inch equal thickness of material.
Let 60,000 pounds per square inch equal tensile strength of material.
Let 40 inches equal diameter of the boiler.
Let 6 equal a factor of safety.
Let 20 per cent. equal amount of pressure to be added for double riveting.

Then we have:

$$\left\{\left(\frac{.25\times 60000}{40\div 2}\right)\div 6\right\}\times 1.2=150 \text{ lbs.}$$ Safe-working pressure for double riveting.

Performing the operation in the ordinary way, we have:

	.25	Thickness of material.
	60000	Tensile strength of material.
Dividing by one-half of the diameter of the boiler. $40\div 2=$ 20)	15000.00	
Factor of safety. 6)	750	
	125 lbs.	Pressure for single riveting.
	1.2	Multiplying by unity and the required percentage annexed is equivalent to adding 20 per cent.
	250	
	125	
	150.0 lbs.	Safe-working pressure for double-riveted longitudinal seams and all rivet holes in the boiler drilled.

SAFE-WORKING PRESSURE OF DOUBLE-RIVETED BOILERS—SIMPLE RULE.

The simplest rule for calculating the safe-working pressure prescribed by the United States laws for double-riveted boilers is as follows:

RULE.—Multiply the thickness of material in the weakest plate in the boiler, in decimals of an inch, by the tensile strength of the material, in pounds per square inch, and divide the product by the diameter of the boiler, in inches, and then multiply the quotient by .4; the product will give the safe-working pressure, per square inch, for a boiler with double-riveted longitudinal seams, if all the rivet holes have been drilled, as required by law.

Example.—Let 25 one hundredths of an inch equal thickness of material.
Let 60,000 pounds per square inch equal tensile strength of material.
Let 40 inches equal diameter of the boiler.
Let .4 equal a constant.

Then we have:

$$\left(\frac{.25 \times 60000}{40}\right) \times .4 = 150 \text{ lbs. Safe-working pressure per square inch.}$$

Performing the operation in a simple way, we have:

$$\begin{array}{rrll}
 & .25 & & \text{Thickness of material.} \\
 & 60000 & & \text{Tensile strength of material.} \\
\hline
\text{Dividing by the diameter of the boiler.}\quad 40) & 15000.00 & (375 & \\
 & 120 & .4 & \text{A constant.} \\
\hline
 & 300 & 150.0 \text{ lbs.} & \text{Safe-working pressure per square inch.} \\
 & 280 & & \\
\hline
 & 200 & & \\
 & 200 & & \\
\hline
\end{array}$$

TENSILE STRENGTH OF BOILER PLATE.

[Section 4430 of the Revised Statutes of the United States; and Section 3, of Rule 1, of the United States Board of Supervising Inspectors of Steam Vessels.]

RULE.—Divide the breaking strain of sample piece, in pounds, by its area of cross section at point of fracture before it is broken, and the quotient will give the tensile strength of the material in pounds per square inch.

Example.—Let 15,525 pounds equal the strain required to pull the sample in two.
Let ·97 one hundredths of an inch equal width of sample at point intended for fracture.
Let 26 one hundredths of an inch equal thickness of sample at point intended for fracture.

Then we have:

$$\frac{15525}{.97 \times .26} = 61558 + \text{lbs. Tensile strength per square inch.}$$

Performing the operation in a simple way, we have:

```
     .97
     .26
    ----
     582
    194
   -----
   .2522  Area of sample piece before breaking
```

Next, dividing the strain at which the sample broke by the area, we have:

```
.2522) 15525.0000 (61558+ lbs. Tensile strength per square inch.
       15132
       -----
         3930
         2522
         -----
         14080
         12610
         -----
          14700
          12610
          -----
           20900
           20176
           -----
```

DUCTILITY OF BOILER PLATE.

[Sections 4430 and 4431 of the Revised Statutes of the United States; and Section 6, of Rule 1, of the United States Board of Supervising Inspectors of Steam Vessels.]

Section 6, of Rule 1, of the Board of Supervising Inspectors, provides that the ductility of boiler plate shall be as follows:

"Iron of 45,000 pounds tensile strength shall show a contraction of area (at point of fracture) of fifteen (15) per cent., each additional 1,000 pounds tensile strength shall show one (1) per cent. additional contraction of area, up to and including 55,000 T. S. Iron of 55,000 T. S. and upward, showing twenty-five (25) per cent. reduction of area, shall be deemed to have the lawful ductility. All steel plate of one-half inch thickness and under, shall show a contraction of area of not less than fifty (50) per cent. Steel plate over one-half inch in thickness up to three-quarters inch in thickness, shall show a reduction of not less than forty-five (45) per cent. All steel plate over three-fourths inch thickness, shall show a reduction of not less than forty (40) per cent."

The terms "contraction of area" and "reduction of area" are synonymous terms, and they are employed interchangeably in the law; and what is meant is the contraction of the material at point of fracture. All iron or steel plate having the quality of being ductile will contract in dimensions at the point at which they are pulled apart; and the extent, amount or per cent. (all of which are the same) of such reduction or contraction of the material at the point at which it is broken, represents its ductility. Iron or steel having no ductility will show no contraction in size of sample at point of fracture. On the other hand, the more ductility the material has the greater will be the contraction and elongation of the sample employed in testing. If the material is very ductile it will stretch considerably before breaking, and consequently reduce considerably at the point where it is pulled apart, while material that has no ductility will not stretch, neither will it be reduced at point of fracture by pulling it apart.

The importance, then, of having boilers made of material of the greatest possible amount of ductility becomes apparent, when it is known that boiler material not having the proper amount of ductility is liable to crack and produce dangerous results. And for that reason no material ought to be allowed in any steam boiler with less ductility than that prescribed by the United States laws; and engineers placed in charge of the building of new boilers should see to it that they are constructed in accordance with these laws in every particular, and more especially for land purposes, where there is no law governing the construction of steam boilers, because, so far as marine boilers are concerned, unless they are made strictly up to the very letter of the law they will be allowed reduced pressure or be condemned.

DUCTILITY OF BOILER PLATE—HOW DETERMINED.

RULE.—First, multiply the width of the sample by its thickness, in decimals of an inch, at the smallest point (the point intended for fracture before breaking) and call the product "the area of sample before breaking."

Second, multiply the width of the sample by its thickness, in decimals of an inch, after breaking at the point of fracture, and call the product "the area of sample after breaking."

Third, subtract "the area of sample after breaking" from "the area of sample before breaking," and then divide the remainder by "the area of sample before breaking," and the quotient will give the percentage of ductility of the material.

Example.—Let 97 one hundredths of an inch equal original width of sample.

Let 26 one hundredths of an inch equal original thickness of sample.

Let 88 one hundredths of an inch equal reduced width of sample.

Let 18 one hundredths of an inch equal reduced thickness of sample.

Then we have:

$$\frac{(.97\times.26)-(.88\times.18)}{.97\times.26}=.37+ \text{ Percentage of ductility.}$$

Performing the operation in the ordinary way, we have:

$.97\times.26=.2522$ Area of sample before breaking.
$.88\times.18=.1584$ Area of sample after breaking.

Dividing the remainder by the area of sample before breaking.

```
.2522 ) .0938.0 ( 0.37+   Percentage of ductility.
         756 6
         -----
         181 40
         176 54
         ------
```

The material, it will be observed, shows a reduction of area or a ductility—which is the same thing in law—of thirty-seven per cent. The material being steel, and having a tensile strength of 60,000 pounds per square inch, the test shows that it is too hard for boiler purposes, and therefore liable to crack when subjected to unequal expansion and contraction, as well as to repeated heating and cooling; and hence it would be condemned by the inspectors for marine purposes. It is, therefore, well to remark here, that any material that is not good enough for marine boilers ought not be allowed in the construction of any other class of boilers. And yet, material condemned by the United States inspectors for use in marine boilers is, with few exceptions, used in the construction of land boilers.

SAFE-WORKING PRESSURE BASED UPON DATA FOR ANY GIVEN MARINE BOILER.

[Section 4433 of the Revised Statutes of the United States; Sections 3 and 6, of Rule 1, and Section 3, of Rule 2, of the United States Board of Supervising Inspectors of Steam Vessels.]

What would be the working pressure allowable, per square inch, in a boiler 48 inches in diameter, made of steel plates 26 one hundredths of an inch in thickness, samples of which, one inch by 26 one hundredths of an inch, broke in testing, at a strain of 15,452 pounds, and reduced in dimensions at point of fracture to 85 one hundredths of an inch in width, and to 15 one hundredths of an inch in thickness; the boiler being made with double-riveted longitudinal seams, and all of the rivet holes in the boiler drilled?

Section 4433 of the Revised Statutes prescribes the working steam pressure allowable, and limits the pressure to produce a strain on the

material in the shell of the boiler not to exceed one-sixth of the tensile strength of the material for single-riveted longitudinal seams, but allows an addition of twenty per cent. for double-riveted longitudinal seams, in case all the rivet holes in the boiler have been drilled.

Section 3, of Rule 1, of the United States Board of Supervising Inspectors relates to the testing of boiler plate, and reads as follows:

"SEC. 3. To ascertain the tensile strength and other qualities of iron plate, there shall be taken from each sheet to be used in shell or other parts of boiler, which are subject to tensile strain, a test piece prepared in form according to the following diagram: 10 inches in length, 2 inches in width—cut out in the center in the manner indicated in Fig. 167.

IRON PLATE.

FORM OF TEST PIECE FOR DETERMINING DUCTILITY AND TENSILE STRENGTH.

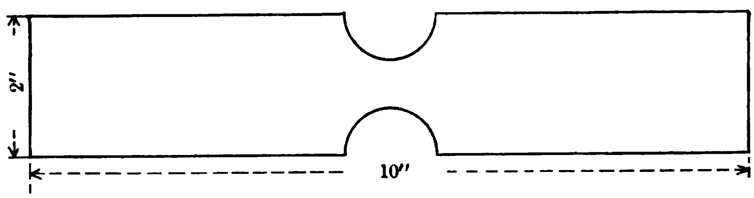


Fig. 167

"All sample pieces of iron plate five-sixteenths ($\frac{5}{16}$) of an inch thick and under shall be one inch wide at reduced section; plate over five-sixteenths ($\frac{5}{16}$) of an inch thick shall be reduced in width at center to an aggregate area approximating four-tenths ($\frac{4}{10}$) of one square inch, but such reduced area shall, in no case, exceed 45 nor be less than 35 one hundredths of an inch, and the force at which the piece can be parted in the direction of the fibre or grain (when of iron) represented in pounds avoirdupois, in proportion to the ratio of its area, shall be deemed the tensile strength per square inch of the plate from which the sample was taken; and should the tensile strength ascertained by the test equal that marked on the plates from which the test pieces were taken, the plates must be allowed to be used in the construction of marine boilers. . . . *Provided, always*, that the plate possesses homogeneousness, toughness, and the ability to withstand the effect of repeated heating and cooling; but should these tests prove any plate to be overstamped (all plates are required to be stamped with the tensile strength of the material), such plate must be rejected as failing to have the strength stamped thereon. But nothing herein

shall be so construed as to prevent the manufacturer from restamping such plate, and all other plates in the lot, at the lowest tensile strength indicated by the deficient sample, provided such restamping is done previous to the use of the plates in the manufacture of marine boilers. When more than one sample shall be tested from one sheet, the sample showing the lowest tensile strength shall be allowed as the tensile strength of the plate."

PERCENTAGE OF DUCTILITY OF BOILER PLATE, TENSILE STRENGTH AND SAFE-WORKING PRESSURE.

The portion of Section 3, of Rule 2, relating to steam pressure allowable for boilers with double-riveted longitudinal seams reads as follows in regard to the drilling of all the rivet holes:

"The pressures allowable on boilers when all the rivet holes have been fairly drilled instead of punched, and the longitudinal laps of their cylindrical parts double riveted."

We will now proceed to answer the question contained at the commencement of this subject, in accordance with the law:

RULE.—First, ascertain the percentage of ductility of the material by subtracting the reduced area of sample test pieces showing the least reduction of area at point of fracture from the original area at same point, and then divide the remainder by the original area, the quotient will give the per cent. of ductility of the material. If a reduction of 50 *per cent. is shown, the material will be allowed, and we proceed with our calculations.*

Second, ascertain the tensile strength of the material by dividing the lowest strain at which any of the samples broke by the area of the sample before breaking, and the quotient will give the tensile strength.

Third, multiply the thickness of the weakest plate in the boiler, in hundredths of an inch, by the tensile strength of the material, in pounds per square inch; then divide the product by one-half of the diameter of the boiler, in inches; then divide the quotient by 6; *then add* 20 *per cent. to the last quotient, and the sum will give the working pressure allowable.*

Example.—Let .85″×.15″ equal dimensions of sample at point of fracture after breaking.
Let 1.00″×.26″ equal dimensions of sample at point of fracture before breaking.
Let 15,452 pounds equal strain at which sample broke.
Let .26″ equal thickness of weakest plate in the boiler.
Let 48″ equal diameter of the boiler.
Let 6 equal a constant.
Let 20 equal per cent. of pressure to be added for double riveting.

Then we have:

$$1.00 \times .26 = .2600 \quad \text{Area of sample before breaking.}$$
$$.85 \times .15 = .1275 \quad \text{Area of sample after breaking.}$$

Dividing remainder by area of sample before breaking.

$$.2600).1325.0\,(0.51\text{— Per cent. ductility.}$$
$$\begin{array}{r} 1300\,0 \\ \hline 25\,00 \\ 26\,00 \\ \hline \end{array}$$

It is now demonstrated that the material possesses the lawful ductility—not less than fifty per cent.—and therefore it will be allowed in the construction of marine boilers. We will therefore next proceed to determine the tensile strength of the material per square inch, in order to determine the working pressure allowable.

Then, we divide the strain at which sample broke by the area of sample at point of fracture before breaking, thus:

$$1.00 \times .26 = .2600)\ 15452.0000\ (59430+ \text{ lbs. Tensile strength per square inch.}$$
$$\begin{array}{r} 13000 \\ \hline 24520 \\ 23400 \\ \hline 11200 \\ 10400 \\ \hline 8000 \\ 7800 \\ \hline 2000 \end{array}$$

Having ascertained the tensile strength of the material per square inch, we now proceed to determine the working pressure allowable under the law, thus:

$$\left(\frac{59430 \times .26}{48 \div 2}\right) \div 6 = 107.31 \text{ lbs. Pressure for single-riveted boiler.}$$

Next we have:

$$\begin{array}{r} 107.31 \\ .20 \\ \hline 21.4620 \end{array} \text{ lbs. 20 per cent. of 107.31.}$$

Then, adding the twenty per cent. to the pressure that would be allowed for single riveting, we have:

$$\begin{array}{rl} 107.31 \text{ lbs.} & \text{Pressure for single riveting.} \\ 21.462 & \text{20 per cent. of 107.31.} \\ \hline 128.772 \text{ lbs.} & \text{Working pressure allowable in the boiler.} \end{array}$$

STEEL PLATE.

FORM OF TEST PIECE FOR DETERMINING ELONGATION AND TENSILE STRENGTH.

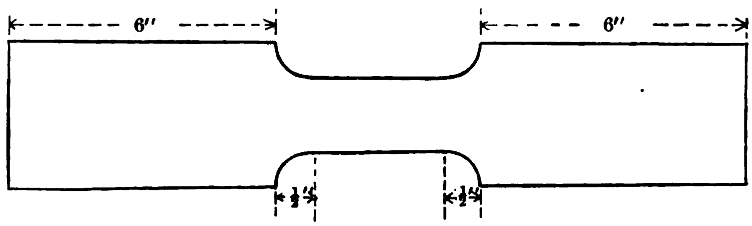


Fig. 168

Section 3, of Rule 1, provides that sample pieces of steel plate, for determining the percentage of elongation, shall be made similar in form to that shown in Fig. 168; but all test pieces one inch, and over one inch, in thickness shall have the straight part of the reduced section 8 inches in length; and that the length of the reduced section of samples having a thickness of less than one inch shall be governed by the following rule:

LENGTH OF REDUCED SECTION OF TEST PIECE.

RULE.—Multiply the width by the thickness, and the product by 8, *the last product will give the required length of reduced width of sample. The width of reduced part is fixed at one inch, and the elongation under the test shall be not less than* 25 *per cent.*

Example.—Let 98 one hundredths of an inch equal width of sample.
Let .30 one hundredths of an inch equal thickness of sample.
Let 8 equal a constant.

Then we have:

$.98 \times .30 \times 8 = 2.3520$ inches. Required length of reduced width of sample.

ELONGATION OF TEST PIECE.

RULE.—Divide the amount of elongation by the original length of the reduced width of test piece, and multiply the quotient by 100, *and the product will give the percentage of elongation.*

Example.—Let 75 one hundredths of an inch equal amount of elongation.

Let 3 inches equal original length of reduced width of test piece.

Let 100 equal a constant.

Then we have:

$$\left(\frac{.75}{3}\right)\times 100=25 \text{ Percentage of elongation.}$$

SPECIAL TEST OF STEEL PLATE.

The last paragraph of Section 3, of Rule 1, provides: "That where contracts for boilers for ocean-going steamers require a test of material in compliance with the British Board of Trade, British Lloyds or Bureau Veritas rules for testing, the inspectors shall make the tests in compliance with the following rules:

"Steel plates shall, in all cases, have an ultimate elongation not less than twenty per cent. in a length of 8 inches. It is to be capable of being bent to a curve of which the inner radius is not greater than one and one-half times the thickness of the plates after having been heated to a low cherry red, and quenched in water of 82° Fahrenheit."

The tests are to be made by United States inspectors, and the test pieces are to be made according to the following form:

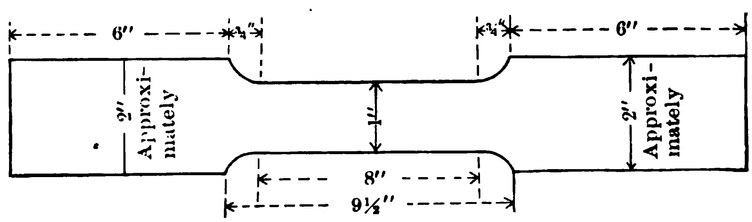


Fig. 169

PERCENTAGE OF ELONGATION.

RULE.—Subtract the length of sample piece before breaking from the length of sample piece after breaking; then divide the remainder by the length of sample piece before breaking, and multiply the quotient by 100; the product will give the per cent. of elongation.

Example.—Let 9.6 inches equal length of sample piece after breaking.

Let 8 inches equal length of sample piece before breaking.

Then we have:

$$\left(\frac{9.6-8}{8}\right)\times 100=20 \text{ Per cent. elongation.}$$

Performing the operation, we have:

$$\begin{array}{r} 9.6 \\ 8 \\ \hline 8)\ 1.6 \\ \hline .2 \\ 100 \\ \hline 20.0 \end{array} \text{ Per cent. elongation.}$$

BUMPED HEADS FOR BOILERS.

WORKING STEAM PRESSURE.

Section 17, of Rule 2, of the United States Board of Supervising Inspectors, provides that the working pressure allowable on bumped heads shall be according to the following rule:

RULE.—Multiply one-sixth of the tensile strength, per square inch, of the plate by the thickness of the plate, and divide the product by six-tenths of the radius to which the head is bumped, which will give the pressure, per square inch, of steam allowed.

Example.—Let 60,000 pounds equal tensile strength of plate per square inch.
Let 6 equal a constant.
Let 25 one hundredths of an inch equal thickness of plate.
Let 24 inches equal radius to which the head is bumped.
Let .6 equal a constant.

Then we have:

$$\frac{(60000\div 6)\times .25}{24\times .6}=173.61+\text{lbs. Pressure per square inch allowable.}$$

Performing the operation, we have:

$$\begin{array}{r} 6)\ 60000 \\ \hline 10000 \\ .25 \\ \hline 50000 \\ 20000 \\ \hline 2500.00 \end{array}$$

Am't carried forward, 2500.00

Dividing by six-tenths of the radius to which the head is bumped.	$24\times.6=14.4)2500.00(173.61+$lbs.	Pressure per square inch allowable.

$$\begin{array}{r} 144 \\ \hline 1060 \\ 1008 \\ \hline 520 \\ 432 \\ \hline 880 \\ 864 \\ \hline 160 \\ 144 \\ \hline \end{array}$$

WORKING STEAM PRESSURE—SIMPLE RULE.

RULE.—First, multiply the thickness of plate, in decimals of an inch, by the tensile strength in pounds per square inch, and call the product "Product No. 1."

Second, multiply the radius, in inches, to which the head is bumped by 6, and multiply the product by .6, and call the last product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the pressure, per square inch, allowable.

Example.—Let 25 one hundredths of an inch equal thickness of plate.
Let 60,000 pounds equal tensile strength of plate per square inch.
Let 24 inches equal radius to which the head is bumped.
Let 6 equal a constant.
Let .6 equal a constant.

Then we have:

$$\frac{.25\times60000}{24\times6\times.6}=173.61+\text{lbs.}$$ Pressure per square inch allowable.

Performing the operation, we have:

.25	Thickness of plate.
60000	Tensile strength of plate.
15000.00	"Product No. 1."

Next we have:

24	Radius to which the head is bumped.
6	A constant.
144	
.6	A constant.
86.4	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
86.4) 15000.00 (173.61+lbs.   Pressure per square inch allowable.
      864
      ----
      6360
      6048
      ----
       3120
       2592
       ----
        5280
        5184
        ----
         960
         864
         ---
```

THICKNESS OF MATERIAL REQUIRED.

RULE.—Multiply the radius, in inches, to which the head is bumped by .6, and multiply the product by the required pressure, in pounds per square inch, and call the last product "Product No. 1."

Second, divide the tensile strength of the plate, in pounds per square inch, by 6, and call the quotient "Quotient No. 1."

Third, divide "Product No. 1" by "Quotient No. 1," and the quotient of this operation will give the required thickness of material in decimals of an inch.

Example.—Let 24 inches equal the radius to which the head is bumped.

Let .6 equal a constant.

Let 173.61 pounds equal the required pressure per square inch.

Let 60,000 pounds equal the tensile strength of plate per square inch.

Let 6 equal a constant.

Then we have:

$$\frac{24\times.6\times173.61}{60000\div6}=.25- \quad \text{Decimals of an inch. Required thickness of plate.}$$

Performing the operation, we have:

```
      24      Radius to which the head is bumped.
      .6      A constant.
    ----
    14.4
  173.61      Required pressure per square inch.
  ------
   69 444
  694 44
 1736 1
 --------
 2499.984     "Product No. 1."
```

Next we have:

```
A constant.  6) 60000  Tensile strength of plate in pounds.
               10000   "Quotient No. 1."
```

Finally, dividing "Product No. 1" by "Quotient No. 1," we have:

```
10000) 2499.984 (.2499+  Decimals of an inch.  Required thickness of plate.
       2000 0
       ------
        499 98
        400 00
        ------
         99 984
         90 000
         ------
          9 9840
          9 0000
          ------
```

THICKNESS OF MATERIAL REQUIRED—SIMPLE RULE.

RULE.—Multiply the required pressure, in pounds per square inch, by the radius, in inches, to which the head is bumped; then multiply the product by the constant 6; *then multiply the product of that operation by* .6; *then divide the last product by the tensile strength, per square inch, of the plate, and the quotient will give the required thickness of material for the head.*

Example.—Let 175 pounds equal required pressure per square inch.
Let 24 inches equal radius to which the head is bumped.
Let 6 equal a constant.
Let .6 equal a constant.
Let 60,000 pounds equal tensile strength of plate per square inch.

Then we have:

$$\frac{175\times24\times6\times.6}{60000}=.252$$ Decimals of an inch. Required thickness of plate.

Performing the operation, we have:

```
                       175
                        24
                      ----
                       700
                      350
                      ----
                      4200
                         6
                     -----
                     25200
                        .6
                   -------
Am't carried forward, 15120.0
```

Dividing by tensile strength of plate per square inch.	60000) 15120.0 (0.252 12000 0 3120 00 3000 00 120 000 120 000	Decimals of an inch. Required thickness of plate.

RADIUS TO WHICH THE HEAD IS REQUIRED TO BE BUMPED.

RULE.—First, multiply the thickness of material for the head, in decimals of an inch, by the tensile strength, in pounds per square inch, and call the product "Product No. 1."

Second, multiply the required pressure, in pounds per square inch, by 6; *then multiply the product by* .6, *and call the last product "Product No. 2."*

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the radius, in inches, to which the head is required to be bumped.

Example.—Let 252 one thousandths of an inch equal thickness of plate.

Let 60,000 pounds equal tensile strength of plate per square inch.

Let 175 pounds equal the required pressure per square inch.

Let 6 equal a constant.

Let .6 equal a constant.

Then we have:

$$\frac{.252\times 60000}{175\times 6\times .6}=24 \text{ inches.}$$ Radius to which the head is required to be bumped.

Performing the operation, we have:

.252	Thickness of plate.
60000	Tensile strength of plate.
15120.000	"Product No. 1."

Next we have:

175	Pressure per square inch.
6	A constant.
1050	
.6	A constant.
630.0	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

630) 15120 (24 inches. 1260 2520 2520	Radius to which the head is required to be bumped.

HYDROSTATIC PRESSURE FOR MARINE BOILERS.

[Section 4418 of the Revised Statutes of the United States.]

The hydrostatic test is made with water, instead of steam pressure, and it is applied to boilers to discover any weakness or defect that could not be discovered by an ocular inspection of the boiler. The boiler is pumped full of water until a certain pressure in the boiler is reached beyond that allowed as a safe-working pressure, and if no defect or weakness is shown, the boiler is considered safe and the working steam pressure prescribed by law is allowed. Section 4418 of the Revised Statutes provides that:

"All boilers used on steam vessels and constructed of iron or steel plates, inspected under the provisions of Section 4430, shall be subjected to a hydrostatic test, in the ratio of one hundred and fifty pounds to the square inch to one hundred pounds to the square inch of the working steam pressure allowed."

HYDROSTATIC PRESSURE FOR SINGLE-RIVETED BOILERS.

RULE.—Multiply the thickness of material, in decimals of an inch, by the tensile strength of the material, in pounds per square inch; then divide the product by one-half the diameter of the boiler, in inches; then divide the quotient by 6; then divide the last quotient by 2, and then add the quotient of the last operation to the preceding quotient, and the sum will give the hydrostatic pressure in pounds per square inch.

Example.—Let 60,000 pounds per square inch equal tensile strength of material.
Let 25 one hundredths of an inch equal thickness of material.
Let 40 inches equal diameter of the boiler.
Let 6 equal a constant.
Let 2 equal a constant.

Then we have:

$$
\begin{array}{rr}
 & .25 \\
 & 60000 \\
\hline
40 \div 2 = 20\,) & 15000.00 \\
\hline
6\,) & 750 \\
\hline
2\,) & 125 \\
 & 62.5 \\
\hline
 & 187.5 \text{ lbs. Hydrostatic pressure.}
\end{array}
$$

HYDROSTATIC PRESSURE FOR SINGLE-RIVETED BOILERS—SIMPLE RULE.

RULE.—Multiply the thickness of material, in hundredths of an inch, by the tensile strength of the material, in pounds per square inch; then divide the product by the diameter of the boiler, in inches; and then multiply the quotient by .5; the last product will give the hydrostatic pressure in pounds per square inch.

Taking the boiler in the previous example, we have:

```
          .25
         60000
     ----------
  40 ) 15000.00 (375
       120         .5
       ---       -----
        300      187.5 lbs. Hydrostatic pressure.
        280
        ---
         200
         200
         ---
```

HYDROSTATIC PRESSURE FOR DOUBLE-RIVETED BOILERS.

RULE.—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength of the material, in pounds per square inch; then divide the product by one-half of the diameter of the boiler, in inches; then divide the quotient by 6; then add 20 per cent. of the last quotient to the quotient; then divide the sum by 2; then add the last quotient to the sum, and the last sum will give the hydrostatic pressure per square inch required.

Example.—Let 25 one hundredths of an inch equal thickness of weakest plate.

Let 60,000 pounds per square inch equal tensile strength of plate.

Let 40 inches equal diameter of the boiler.

Let 6 equal a constant.

Let 20 per cent. equal additional safe-working pressure for double riveting.

Let 2 equal a constant.

Then we have:

```
                  .25     Thickness of weakest plate.
                 60000    Tensile strength of material.
               ---------
  40÷2=   20) 15000.00
               ---------
             6) 750
               ----
*Am't carried forward,*   125 lbs. Working pressure for single riveting.
```

Am't brought forward,	125 lbs.	Working pressure for single riveting.
Adding 20% $125 \times .20 =$	25 lbs.	Twenty per cent. of working pressure.
	2) 150 lbs.	Working pressure for double riveting.
	75 lbs.	One-half of the working pressure.
	225 lbs.	Hydrostatic pressure required.

HYDROSTATIC PRESSURE FOR DOUBLE-RIVETED BOILERS—SIMPLE RULE.

RULE.—Multiply the thickness of material in the weakest plate, in decimals of an inch, by the tensile strength of the material, in pounds per square inch; then divide the product by the diameter of the boiler, in inches; then multiply the quotient by .6; the last product will give the hydrostatic pressure in pounds per square inch.

Taking the boiler employed in the previous example, we have:

$$\left(\frac{.25 \times 60000}{40}\right) \times .6 = 225 \text{ lbs.}$$ Hydrostatic pressure per square inch.

Performing the operation in the ordinary way, we have:

```
                                     .25      Thickness of material.
                                   60000      Tensile strength of material.
Dividing by diameter of boiler. 40) 15000.00 (375
                                   120           .6  A constant for double riveting.
                                    300      225.0 lbs.  Hydrostatic pressure per square inch.
                                    280
                                     200
                                     200
```

HYDROSTATIC PRESSURE FOR SINGLE AND DOUBLE RIVETED BOILERS.

The rule for determining the hydrostatic pressure for both single and double riveted longitudinal seams in condensed form is as follows:

RULE.—Add one-half of the working pressure allowable to the working pressure, and the sum will give the hydrostatic pressure required.

Example.—Let 150 pounds per square inch equal the working pressure for either single or double riveted longitudinal seams.

Then we have:

```
2) 150
    75
   225 lbs.  Hydrostatic pressure required.
```

REDUCTION OF WORKING STEAM PRESSURE.

[Section 4418 of the Revised Statutes of the United States.]

Under authority of Section 4418 of the Revised Statutes, inspectors may at any time cut down the working steam pressure to the limit of safety, for any deterioration in the material, or other causes of weakness produced by the use of the boiler for a long period, or on account of negligence on the part of the engineer in failing to take proper care of the boiler, and thus causing weakness or deterioration in the material. Hence, at inspections a lower hydrostatic pressure is applied than would be in the case of a new boiler, and the working steam pressure allowed is always two-thirds of the hydrostatic pressure. Therefore, while in theory the working steam pressure is based upon the strength of the material in the boiler, and the manner in which the boiler is constructed, yet in practice the working steam pressure is based upon the hydrostatic pressure that has been applied, and this hydrostatic pressure is always based upon the strength of the boiler.

THICKNESS OF MATERIAL FOR MARINE BOILERS LIMITED.

[Section 4434 of the Revised Statutes of the United States.]

This section of the Revised Statutes of the United States limits the thickness of material in the shell of boilers, to which the heat is applied to the outside of the shell, to 30 ($\frac{30}{100}$) one hundredths of an inch for all steamers on the Mississippi River and its tributaries, and to boilers on all other steamers to one-half inch.

THICKNESS OF MATERIAL REQUIRED FOR SINGLE-RIVETED BOILERS.

RULE.—Multiply the given pressure, in pounds per square inch, by 6; then multiply the product by one-half of the diameter of the boiler, in inches; then divide the last product by the tensile strength of the material, in pounds per square inch, and the quotient will give the thickness of material required in decimals of an inch.

Example.—Let 125 pounds per square inch equal the given pressure.
Let 6 equal a constant.
Let 40 inches equal diameter of the boiler.
Let 60,000 pounds per square inch equal tensile strength of material.

Then we have:

$$\frac{125\times6\times(40\div2)}{60000}=.25 \text{ inches. Thickness of material required.}$$

Performing the operation in the ordinary way, we have:

		125	Working steam pressure.
		6	A constant.
		750	
Multiplying by one-half of diameter of boiler.	40÷2=	20	
Dividing by tensile strength of material.	60000)	15000.0 (0.25	Decimals of an inch. Thickness of material required.
		12000 0	
		3000 00	
		3000 00	

THE WORKING STEAM PRESSURE BASED ON THE HYDROSTATIC PRESSURE.

[Section 4418 of the Revised Statutes of the United States.]

In no case will a working steam pressure per square inch be allowed greater than two-thirds of the hydrostatic pressure that has been applied. It sometimes happens that the required amount of hydrostatic pressure can not be obtained on account of leakages in valves and joints; and even in such cases only two-thirds of the hydrostatic pressure obtained will be allowed as a working steam pressure. If the owner of the boiler desires the full amount of steam pressure to which his boiler would otherwise be entitled, he must get his valves and joints tight enough to withstand the increased pressure that the hydrostatic pressure will produce, and then the hydrostatic pressure must again be applied, and if the required pressure in making the hydrostatic test is obtained the boiler will be allowed the amount of steam pressure to which it would otherwise be entitled, but not until then.

UNITED STATES TABLE OF PRESSURES ALLOWABLE ON MARINE BOILERS.

Diameter of Boiler.	Thickness of Plates.	45,000 Tensile Strength. 1-6 7,500		50,000 Tensile Strength. 1-6 8,333.3		55,000 Tensile Strength. 1-6 9,166.6		60,000 Tensile Strength. 1-6 10,000		65,000 Tensile Strength. 1-6 10,833.3		70,000 Tensile Strength. 1-6 11,666.6	
		Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l
36 INCHES	.1875	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82
	.21	87.5	105.00	97.21	116.65	106.94	128.3	116.66	139.99	126.38	151.65	136.11	163.33
	.23	95.83	114.99	106.47	127.76	117.12	140.54	127.77	153.32	138.41	166.09	149.07	178.88
	.25	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43
	.26	108.33	129.9	120.37	144.44	132.4	158.88	144.44	173.32	156.48	187.77	168.51	202.21
	.29	120.83	144.99	134.25	161.11	147.68	177.21	161.11	193.33	174.53	209.43	187.90	225.48
	.3125	130.2	156.24	144.67	173.6	159.14	190.96	173.6	208.32	188.07	225.68	202.5	243.04
	.33	137.5	165.00	152.77	183.32	168.05	201.66	183.33	219.99	198.61	238.33	213.88	256.65
	.35	145.83	174.99	162.03	194.43	178.23	213.87	194.44	233.32	210.64	252.76	226.84	272.20
	.375	156.25	187.5	173.61	208.33	190.97	229.16	208.33	249.99	225.69	271.82	243.05	291.66
38 INCHES	.1875	74.01	88.89	82.23	98.67	90.46	108.54	98.68	118.41	106.9	128.28	115.13	138.16
	.21	82.89	99.46	92.1	110.52	101.31	121.57	110.52	132.62	119.73	143.67	128.93	154.71
	.23	90.78	108.93	100.87	121.04	110.96	133.15	121.05	145.26	131.13	157.35	141.22	169.46
	.25	98.68	118.41	109.64	131.56	120.61	144.73	131.57	157.88	142.54	171.04	153.5	184.20
	.26	102.63	123.15	114.03	136.83	125.43	150.51	136.84	164.2	148.24	177.88	159.64	191.56
	.29	114.47	137.36	127.19	152.62	139.91	167.89	152.63	183.15	165.35	198.42	178.06	213.67
	.3125	123.35	148.02	137.00	164.46	150.76	180.91	164.47	197.36	178.17	213.08	191.88	230.25
	.33	130.26	156.31	144.73	173.67	159.2	191.04	173.68	208.41	188.15	225.78	202.62	243.14
	.35	138.15	165.78	153.5	184.21	168.85	202.62	184.21	221.05	199.56	239.47	214.91	257.89
	.375	148.00	177.60	164.73	197.67	180.91	217.09	197.36	236.83	213.81	256.57	230.26	276.31
40 INCHES	.1875	70.31	84.37	78.12	93.74	85.93	103.11	93.75	112.5	101.56	121.87	109.37	131.24
	.21	78.75	94.50	87.49	104.98	96.24	115.48	105.00	126.00	113.74	136.48	122.49	146.98
	.23	86.25	103.5	95.83	114.99	105.41	126.49	115.00	138.00	124.58	149.49	134.16	160.99
	.25	93.75	112.5	104.16	124.99	114.58	137.49	125.00	150.00	135.41	162.49	145.83	174.99
	.26	97.5	117.00	108.33	129.99	119.16	142.99	130.00	156.00	140.83	168.99	151.66	181.99
	.29	108.75	130.5	120.83	144.99	132.91	159.49	145.00	174.00	157.08	188.49	169.16	202.99
	.3125	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.45	169.27	203.12	182.29	218.74
	.33	123.75	148.5	137.49	164.98	151.24	181.48	165.00	198.00	178.74	214.48	192.49	230.98
	.35	131.25	157.5	145.83	174.99	160.41	192.49	175.00	210.00	189.58	227.49	204.16	244.99
	.375	140.62	168.74	156.24	187.48	171.87	206.24	187.5	225.00	203.12	243.74	218.74	262.48

UNITED STATES TABLE OF PRESSURES ALLOWABLE ON MARINE BOILERS.—Continued.

Diameter of Boiler.	Thickness of Plates.	45,000 Tensile Strength. 1-6 7,500		50,000 Tensile Strength. 1-6 8,333.3		55,000 Tensile Strength. 1-6 9,166.6		60,000 Tensile Strengt 1-6 10,000		65,000 Tensile Strength. 1-6 10,833.3		70,000 Tensile Strength. 1-6 11,666.6	
		Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l
	.1875	66.96	80.35	74.40	89.28	81.84	98.20	89.28	107.13	96.72	116.06	104.16	124.99
	.21	75.00	90.00	83.32	99.99	91.66	109.99	100.00	120.00	108.33	129.99	116.66	139.99
	.23	82.14	98.56	91.23	105.51	100.39	120.46	109.52	131.42	118.65	142.38	127.77	153.32
42	.25	89.28	107.13	99.2	119.04	109.12	130.94	119.04	142.84	128.96	154.75	138.88	166.65
	.26	92.85	111.42	103.17	123.8	113.49	136.18	123.8	148.56	134.12	160.94	144.44	173.32
INCHES	.29	103.57	124.28	115.07	138.08	126.57	151.85	138.09	165.7	149.6	179.52	161.11	193.33
	.3125	111.6	133.92	124.00	148.8	136.4	163.68	148.74	178.56	161.2	193.44	173.61	208.23
	.33	117.85	141.42	130.94	157.12	144.04	172.84	157.14	188.56	170.23	204.27	183.33	219.99
	.35	125.00	150.00	138.88	166.65	152.77	183.32	166.66	199.99	180.55	216.66	194.44	233.32
	.375	133.92	160.7	148.8	178.56	163.68	196.40	178.57	214.28	193.45	232.14	208.33	249.99
	.1875	63.92	76.7	71.02	85.22	78.12	93.74	85.22	102.26	92.32	110.78	99.42	119.3
	.21	71.59	85.9	79.54	95.44	87.49	104.98	95.45	114.54	103.4	124.08	111.36	133.63
	.23	78.4	94.08	87.12	104.54	95.83	114.99	104.54	125.44	113.25	135.9	121.96	146.35
44	.25	85.22	102.26	94.69	113.62	104.16	124.99	113.63	136.35	123.1	147.72	123.56	159.07
	.26	88.63	106.35	98.48	118.17	108.33	129.99	118.18	141.81	128.02	153.62	137.87	165.44
INCHES	.29	98.86	118.63	109.84	131.80	120.83	144.99	131.81	158.17	142.79	171.33	153.78	184.53
	.3125	106.53	127.83	118.36	142.03	130.2	156.24	142.04	170.44	153.88	184.65	165.71	198.85
	.33	112.5	135.00	124.99	149.98	137.49	164.98	150.00	180.00	162.49	194.98	174.99	209.98
	.35	119.31	143.17	132.57	159.08	145.83	174.99	159.09	190.9	172.34	206.8	185.6	222.72
	.375	127.81	153.37	142.04	170.44	156.24	187.48	170.45	204.54	184.65	221.58	198.86	238.63
	.1875	61.14	73.36	67.93	81.51	74.72	89.66	81.51	97.81	88.31	105.97	95.1	114.12
	.21	68.47	82.16	76.08	91.29	83.69	100.42	91.3	109.56	98.91	118.69	106.52	127.82
	.23	75.00	90.00	83.33	100.00	91.66	109.99	100.00	120.00	108.33	129.99	116.66	139.99
46	.25	81.51	97.81	90.57	108.68	99.63	119.55	108.69	130.42	117.75	141.3	126.8	152.16
	.26	84.78	101.73	94.2	113.04	103.62	124.34	113.44	135.64	122.46	146.95	131.88	158.25
INCHES	.29	94.56	113.47	105.07	126.00	115.57	138.68	126.09	151.3	136.59	163.92	147.1	176.52
	.3125	101.9	122.28	113.21	135.85	124.54	149.44	135.86	163.03	147.19	176.62	158.51	190.21
	.33	107.6	129.12	119.56	143.47	131.52	157.82	143.97	172.16	155.43	186.51	167.39	200.86
	.35	114.13	136.95	126.8	152.16	139.49	167.38	152.17	182.6	164.85	197.82	177.53	213.03
	.375	122.28	146.73	135.86	163.03	149.45	179.34	163.04	195.64	176.62	211.94	190.21	228.25

UNITED STATES TABLE OF PRESSURES ALLOWABLE ON MARINE BOILERS.—Continued.

Diameter of Boiler.	Thickness of Plates.	45,000 Tensile Strength. 1-6 7,500 Pressure.	45,000 20 per ct. Addition'l	50,000 Tensile Strength. 1-6 8,333.3 Pressure.	50,000 20 per ct. Addition'l	55,000 Tensile Strength. 1-6 9,166.6 Pressure.	55,000 20 per ct. Addition'l	60,000 Tensile Strength. 1-6 10,000 Pressure.	60,000 20 per ct. Addition'l	65,000 Tensile Strength. 1-6 10,833.3 Pressure.	65,000 20 per ct. Addition'l	70,000 Tensile Strength. 1-6 11,666.6 Pressure.	70,000 20 per ct. Addition'l
48 INCHES	.1875	58.59	70.30	65.1	78.12	71.61	85.93	78.12	93.74	84.63	101.55	91.13	109.35
	.21	65.62	78.74	72.91	87.49	80.2	96.24	87.49	104.98	94.79	113.74	102.08	122.49
	.23	71.87	86.24	79.85	95.82	87.84	105.4	95.83	114.99	103.81	124.57	111.8	133.16
	.25	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82
	.26	81.25	97.50	90.27	108.32	99.3	119.16	108.33	129.99	117.36	140.83	126.38	151.65
	.29	90.62	108.74	100.69	120.82	110.76	132.91	120.83	144.99	130.9	157.08	140.97	169.16
	.3125	97.65	117.18	108.56	130.2	119.35	143.22	130.21	156.25	141.05	169.26	151.9	182.28
	.33	103.12	123.74	114.58	137.49	126.04	151.24	137.5	165.00	148.95	178.74	160.41	192.49
	.35	109.37	131.24	121.52	145.82	132.67	160.4	145.83	174.99	157.98	189.58	170.13	204.15
	.375	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.50	169.27	203.12	182.29	218.74
54 INCHES	.1875	52.08	62.49	57.87	69.44	63.65	76.38	69.44	82.44	75.23	90.27	81.01	97.21
	.21	58.33	69.99	64.81	77.77	71.29	85.54	77.77	93.32	84.25	101.1	90.74	108.88
	.23	63.88	76.65	70.98	85.17	78.08	93.69	85.18	102.21	92.28	110.73	99.38	119.25
	.25	69.44	83.32	77.16	92.59	84.87	101.84	92.59	111.10	100.3	120.36	108.02	129.62
	.26	72.22	86.66	80.24	96.28	88.27	105.92	96.29	115.54	104.31	125.17	112.44	134.8
	.29	80.55	96.66	89.5	107.40	98.45	118.14	107.41	128.88	116.35	139.62	125.3	150.36
	.3125	86.8	104.16	96.44	115.72	106.09	127.30	115.55	138.66	125.38	150.45	135.03	162.03
	.33	91.66	109.99	101.84	122.22	112.03	134.43	122.22	146.66	132.4	158.88	142.59	171.10
	.35	97.22	116.66	108.02	129.62	118.82	142.58	129.62	155.54	140.43	168.51	151.23	181.47
	.375	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43
60 INCHES	.1875	46.87	56.24	52.08	62.49	57.29	68.74	62.5	75.00	67.7	81.24	72.91	87.49
	.21	52.5	63.00	58.33	69.99	64.16	76.99	69.99	84.00	75.83	90.99	81.66	97.99
	.23	57.5	69.00	63.88	76.65	70.27	84.32	76.66	91.99	83.05	99.66	89.44	107.32
	.25	62.5	75.00	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66
	.26	65.00	78.00	72.22	86.66	79.44	95.32	86.66	103.99	93.88	112.65	101.11	121.33
	.29	72.5	87.00	80.55	96.66	88.61	106.33	96.66	115.99	104.72	125.66	112.77	135.32
	.3125	78.12	93.74	86.8	104.16	95.48	114.57	104.18	124.99	112.95	135.54	121.52	145.82
	.33	82.5	99.00	91.66	109.99	100.83	120.99	109.99	132.00	119.16	142.99	128.33	153.99
	.35	87.5	105.00	97.22	116.66	106.94	128.32	116.66	133.99	126.38	151.65	136.11	163.33
	.375	93.75	112.5	104.16	124.99	114.58	137.49	125.00	150.00	135.41	162.49	145.83	174.99

UNITED STATES TABLE OF PRESSURES ALLOWABLE ON MARINE BOILERS.—Continued.

Diameter of Boiler.	Thickness of Plates.	45,000 Tensile Strength. 1-6 7,500		50,000 Tensile Strength. 1-6 8,333 3		55,000 Tensile Strength. 1-6 9,166.6		60,000 Tensile Strength. 1-6 10.000		65,000 Tensile Strength. 1-6 10,833.3		70,000 Tensile Strength. 1-6 11,666.6	
		Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l
66 INCHES	.1875	42.61	51.13	47.34	56.8	52.07	62.49	56.81	68.17	61.55	73.86	66.28	79.53
	.21	47.72	57.26	53.00	63.63	58.33	69.99	63.63	76.35	68.93	82.71	74.24	89.08
	.23	52.27	62.72	58.00	69.69	63.88	76.65	69.69	83.62	75.5	90.6	81.31	97.57
	.25	56.81	68.17	63.13	75.75	69.44	83.32	75.75	90.90	82.07	98.48	88.37	106.04
	.26	59.09	70.9	65.65	78.78	72.22	86.66	78.78	94.53	85.35	102.42	91.91	110.29
	.29	65.90	79.08	73.23	87.87	80.55	96.66	87.87	105.44	95.2	114.24	102.52	123.02
	.3125	71.00	85.2	78.91	94.69	86.89	104.16	94.69	113.62	102.58	123.09	110.47	132.56
	.33	75.00	90.00	83.33	99.99	91.66	109.99	99.99	120.00	108.33	129.99	116.66	139.99
	.35	79.56	95.47	88.38	106.05	97.22	116.66	106.00	127.27	114.89	137.86	123.73	148.47
	.375	85.22	102.26	94.69	113.62	104.16	124.99	113.62	136.34	123.1	147.72	132.57	159.08
72 INCHES	.1875	39.06	46.87	43.4	52.08	47.74	57.28	52.08	62.49	56.42	67.70	60.76	72.91
	.21	43.75	52.5	48.6	58.33	53.47	64.16	58.33	69.99	63.19	75.82	68.05	81.66
	.23	47.91	57.49	53.24	63.88	58.56	70.27	63.88	76.65	62.21	83.05	74.53	89.43
	.25	52.08	62.49	57.87	69.44	63.65	76.38	69.44	83.32	75.22	90.26	81.01	97.21
	.26	54.16	64.99	60.18	72.21	66.2	79.44	72.22	86.66	78.24	93.88	84.25	101.10
	.29	60.41	72.49	67.12	80.54	73.84	88.60	80.55	96.66	87.26	104.71	93.98	112.77
	.3125	65.10	78.12	72.33	86.8	79.57	95.48	86.8	104.16	94.03	112.83	101.27	121.52
	.33	68.75	82.5	76.38	91.65	84.02	100.82	91.66	109.99	99.3	119.16	106.94	128.32
	.35	72.91	87.49	81.01	97.21	89.11	106.93	97.22	116.66	105.32	126.38	113.42	136.1
	.375	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.43	121.52	145.82
78 INCHES	.1875	36.05	43.21	40.06	48.07	44.07	52.87	48.07	57.68	52.08	62.49	56.08	67.29
	.21	40.38	48.45	44.87	53.84	49.35	59.22	53.84	64.60	58.33	69.99	62.82	75.38
	.23	44.23	53.07	49.14	58.96	54.05	64.86	58.95	70.76	63.88	76.65	68.80	82.56
	.25	48.07	57.68	53.41	64.09	58.76	70.5	64.4	76.92	69.44	83.32	74.78	89.73
	.26	50.00	60.00	55.55	66.66	66.11	73.33	66.66	79.99	72.22	86.66	77.77	93.32
	.29	55.76	66.91	61.96	74.35	68.16	81.79	74.35	89.22	80.55	96.66	86.75	104.1
	.3125	60.09	72.1	66.77	80.12	73.45	88.14	80.12	96.14	86.8	104.16	93.48	112.17
	.33	63.46	76.15	70.51	84.61	77.56	93.07	84.61	101.53	91.66	109.99	98.71	118.45
	.35	67.3	80.76	74.78	89.73	82.26	98.71	89.74	107.68	97.22	116.66	104.70	125.64
	.375	72.11	86.53	80.12	96.14	88.14	105.76	96.15	115.38	104.16	124.99	112.17	134.6

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UNITED STATES TABLE OF PRESSURES ALLOWABLE ON MARINE BOILERS.—Continued.

Diameter of Boiler.	Thickness of Plates.	45,000 Tensile Strength. 1-6 7,500		50,000 Tensile Strength. 1-6 8,333.8		55,000 Tensile Strength. 1-6 9,166.6		60,000 Tensile Strength. 1-6 10,000		65,000 Tensile Strength. 1-6 10,833.3		70,000 Tensile Strength. 1-6 11,666.6	
		Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l	Pressure.	20 per ct. Addition'l
84 INCHES	.1875	33.48	40.17	37.2	44.68	40.92	49.1	44.64	53.56	48.36	58.03	52.08	62.49
	.21	37.5	45.00	41.66	49.99	45.83	54.99	50.00	60.00	54.16	64.99	58.33	69.99
	.23	41.02	49.22	45.63	54.75	50.19	60.22	54.75	65.71	59.32	71.18	63.65	76.38
	.25	44.64	53.56	49.6	59.52	54.56	65.47	59.52	71.42	64.48	77.37	69.44	83.32
	.26	46.42	55.7	51.58	61.89	56.74	68.08	61.9	74.28	67.05	80.46	72.22	86.66
	.29	51.78	62.13	57.53	69.03	63.29	75.94	69.04	82.84	74.8	89.76	80.55	96.66
	.3125	55.8	66.96	62.00	74.4	68.2	81.84	74.4	89.28	80.6	96.72	86.8	104.16
	.33	58.92	70.7	65.47	78.56	72.02	86.42	78.57	94.28	85.11	102.13	91.66	109.99
	.35	62.5	75.00	60.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66
	.375	66.96	80.35	74.4	89.28	81.84	98.2	89.28	107.13	96.72	116.06	104.16	124.99
90 INCHES	.1875	31.25	37.5	34.72	41.66	38.19	45.82	41.66	49.99	45.13	54.15	48.68	58.33
	.21	35.00	42.00	38.88	46.65	42.77	51.32	46.66	55.99	50.55	60.66	54.44	65.32
	.23	38.33	45.99	42.59	51.10	46.85	56.22	51.11	61.33	55.37	66.44	59.62	71.54
	.25	41.66	49.99	46.29	55.54	50.92	61.1	55.55	66.66	60.18	72.21	64.81	77.77
	.26	43.33	51.99	48.14	57.76	52.96	63.55	57.77	69.32	62.59	75.1	67.4	80.88
	.29	48.33	57.99	53.7	64.44	59.07	70.8	64.44	77.32	69.81	83.77	75.18	90.21
	.3125	52.08	62.49	57.86	69.43	63.65	76.38	69.44	83.32	75.23	90.27	81.01	97.21
	.33	55.00	66.00	61.11	73.33	67.22	80.66	73.33	87.99	79.44	95.32	85.55	102.66
	.35	58.33	69.99	64.81	77.77	71.29	85.54	77.77	93.32	84.25	101.1	90.72	108.88
	.375	62.5	75.00	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66
96 INCHES	.1875	29.29	35.14	32.55	39.06	35.8	42.96	39.06	46.87	42.31	50.77	45.57	54.68
	.21	32.81	39.37	36.45	43.74	40.1	48.12	43.75	52.5	47.39	56.86	51.04	61.24
	.23	35.93	43.11	39.93	47.91	43.92	52.7	47.91	57.49	51.9	62.28	55.9	67.08
	.25	39.06	46.87	43.4	52.08	47.74	57.28	52.08	62.49	56.42	67.67	60.76	72.91
	.26	40.62	48.74	45.14	54.16	49.65	59.58	54.16	64.99	58.78	70.53	63.19	75.82
	.29	45.31	54.37	50.34	60.4	55.38	66.45	60.41	72.49	65.45	78.54	70.48	84.57
	.3125	48.82	58.58	54.25	65.1	59.67	71.6	65.1	78.12	70.52	84.62	75.95	91.14
	.33	51.56	61.87	57.29	68.74	63.02	75.62	68.75	82.5	74.47	89.36	80.2	96.24
	.35	54.68	65.61	60.76	72.91	66.83	80.19	72.91	87.49	78.99	94.78	85.06	102.07
	.375	58.58	70.29	65.1	78.12	71.61	85.93	78.12	93.74	84.63	101.55	91.14	109.6

FLAT SURFACES IN STEAM BOILERS.

Sections 6 and 7 of Rule 2, of the United States Board of Supervising Inspectors of Steam Vessels, regulates the thickness of boiler plate for flat surfaces, diameters of stay bolts, pitch of stay bolts, and distances between braces, the kind of stay bolts and braces, and manner of attaching the same, the manner of testing steel bars intended for use as stay bolts, and the steam pressure allowable for flat surfaces.

ENGINEERS REQUIRED TO CALCULATE STRAINS ON STAYS AND BRACES.

Not only are inspectors required to thoroughly understand the rules for making the various calculations to enable them to determine the steam pressure allowable in all cases, but, under Section 5, of Rule 5, of the Board of Supervising Inspectors, no person can obtain a license as an engineer, or have the grade of his license raised, unless he can mathematically determine the strain to which stay bolts and braces are subjected by both steam and hydrostatic pressure.

The lack of information on this important subject, on the part of many engineers and boiler makers, has been a most fruitful source of serious steam boiler accidents. It is, therefore, of vital importance that those who build, as well as those under whose care and management they are placed, should fully understand the rules governing the construction of flat surfaces and steam pressure allowable.

AREA OF SURFACE STAYED BY BOLT.

Before entering upon a study of the rules governing the construction of flat surfaces, it will be well for the student to familiarize himself with the rules which will enable him to determine the extent or area of surface stayed by any given bolt or stay. His attention is therefore invited to a careful study of the following rules and illustrations:

RULE.—Multiply the distance, in inches, from center to center of bolts by itself, or, in other words, square the distance of bolts from center to center, and the product will give the number of square inches contained in the area of surface stayed by the bolt or stay.

Example.—Let 5 inches equal distance from center to center of stay bolts.

Then we have:

$5 \times 5 = 25$ square inches. Area of surface to be stayed by each bolt.

To illustrate this more clearly, we will take a section from the flat surface of a boiler, as shown in Fig. 170.

It will be observed that whatever pressure per square inch is put upon the section, as shown in Fig. 170, the load is distributed equally

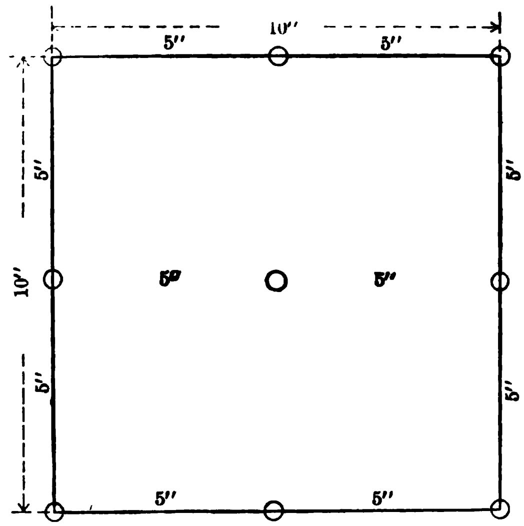


Fig. 170

between the center and the outer bolts, and the natural conclusion would be that the center bolt bears one-half of the entire strain put upon the whole surface 10"×10", as shown in the diagram. But that is not true, as the center bolt bears but one-fourth of the entire strain, as shown in Fig. 171.

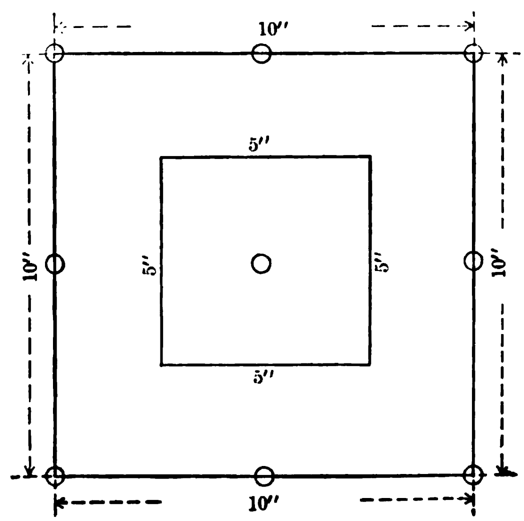


Fig. 171

It will be observed that the width and breadth of the inner diagram, as shown in Fig. 171, are just one-half of those of the outer diagram, while it has an area of but one-fourth of that of the entire diagram, hence it bears but one-fourth of the entire strain; it therefore follows that the center bolt bears but one-fourth of the entire strain, and the other bolts combined bear three-fourths of the strain on that particular section. The question now arises, how much of the entire strain does each of the outer bolts bear? This is illustrated in Fig. 172:

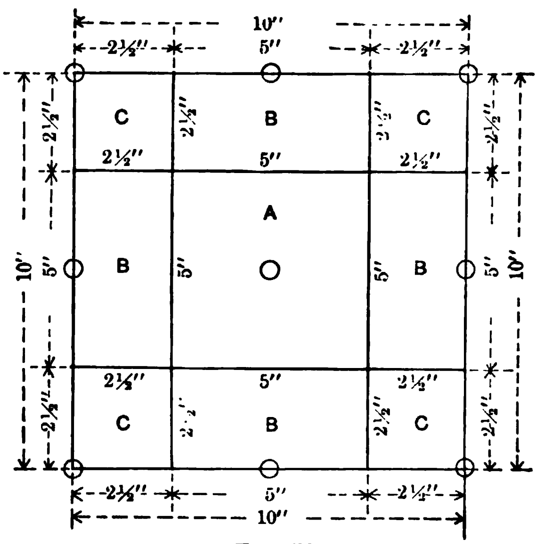


Fig. 172

With a pressure of 100 pounds on the square inch, there would be a total pressure on the entire surface of the diagram, as shown in Fig. 172, of 10,000 pounds, including the space occupied by the bolts or parts of bolts within the diagram, which space should be deducted to obtain perfect accuracy, for the reason that there is no strain or pressure on the space occupied by the bolts.

The area of the entire surface is $10'' \times 10'' = 100$ square inches, and that multiplied by 100 pounds, pressure per square inch, gives the total strain on the entire surface: $100 \times 100 = 10{,}000$ pounds; and this divided by 4, gives the strain borne by the center bolt: $10{,}000 \div 4 = 2500$ pounds, total strain on A in the diagram.

Diagrams B B B B, as shown in Fig. 172, are each but one-halt the size of A, and C C C C are each but one-half the size of B, or one-fourth the size of A. Therefore, it will be observed, that B B B B,

each having but one-eighth of the entire surface, each bears but one-eighth of the entire pressure, and that the four combined bear one-half of the entire pressure.

C C C C, each containing but one-sixteenth of the entire surface, each bears but one-sixteenth of the entire pressure, and consequently each corner bolt bears but one-sixteenth of the entire strain, while each outer center bolt bears one-eighth of the entire strain.

To secure perfect accuracy in determining the amount of strain produced by a given pressure per square inch on the center bolt, the area of cross section of the bolt must be deducted from the area of diagram A, while one-half of the area of the cross section of the bolts in diagrams B B B B, and one-fourth of the area of the cross section of the bolts in diagrams C C C C, must be deducted, as these proportions represent the amount of area of the bolts embraced in their respective diagrams, as will be seen by reference to Fig. 172.

CONSTRUCTION OF FLAT SURFACES AND STEAM PRESSURE ALLOWABLE. PLATES NOT OVER 7-16 OF AN INCH THICK.

STRAIN ON BOLTS LIMITED TO 6000 POUNDS PER SQUARE INCH OF SECTION.

Furnaces, fire boxes, back connections and other flat surfaces; screw stay bolts and nuts, or plain bolts with single nut and socket, or riveted head and socket, or screw stay bolts, ends riveted, pitch limited to 10½ inches; screw stay bolts without socket, not allowed where salt water is used in boilers.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 6000, and call the answer "The Quotient."

Second, divide "The Quotient" by .7854, and extract the square root of the last quotient; the answer will give the required diameter of stay bolt at the bottom of thread in decimals of an inch.

Example.—Let 165.67 pounds equal given steam pressure per square inch.

Let 4 inches equal pitch of stay bolts from center to center.

Let 6000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{165.67\times4\times4}{6000}\right)\div.7854}=.75+$$ Decimals of an inch. Diameter of bolt at bottom of thread.

Performing the operation in the ordinary way, we have:

```
                                 165.67   Given pressure per square inch.
                       4×4=          16   Square of distance between centers of bolts.
                                 ------
                                 994 02
                                1656 7
                                -------
Dividing by the constant. 6000) 2650.72 (0.44178666+   "The Quotient."
                                2400 0
                                -------
                                 250 72
                                 240 00
                                 -------
                                  10 720
                                   6 000
                                  -------
                                   4 7200
                                   4 2000
                                   -------
                                    52000
                                    48000
                                    -------
                                     40000
                                     36000
                                     ------
                                      40000
                                      36000
                                      ------
                                       40000
                                       36000
                                       -----
```

Next, dividing "The Quotient" by .7854, we have:

```
.7854) .44178666 (.5625—   The last quotient.
        39270
        -----
         49086
         47124
         -----
          19626
          15708
          -----
           39186
           39270
           -----
```

Finally, extracting the square root of the last quotient, we have:

```
     .5625 (.75   Decimals of an inch. Diameter of bolt
      49                 at bottom of thread.
     ----
145) 725
     725
     ---
```

PITCH OF BOLTS BASED ON DIAMETER OF BOLTS AND STEAM PRESSURE.

RULE.—First, multiply the area of cross section of the stay bolt at the bottom of thread by 6000, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the required distance, in inches, from center to center of stay bolts.*

Example.—Let 75 one hundredths of an inch equal the diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 165.67 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{.75\times.75\times.7854\times6000}{165.67}}=4 \text{ inches.} \quad \text{Distance of bolts from center to center.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl}
.75 & \text{Diameter of bolt.} \\
.75 & \text{Diameter of bolt.} \\
\hline
375 & \\
525 & \\
\hline
.5625 & \text{Square of diameter of bolt.} \\
.7854 & \text{A constant.} \\
\hline
22500 & \\
28125 & \\
45000 & \\
39375 & \\
\hline
.44178750 & \text{Area of cross section of bolt at bottom of thread.} \\
6000 & \text{A constant.} \\
\hline
2650.72500000 & \text{"Product No. 1."}
\end{array}$$

Next, dividing "Product No. 1" by the given steam pressure (165.67 pounds per square inch), we have:

$$\begin{array}{r}
165.670)\,2650.725\,(16+ \quad \text{The quotient.} \\
1656\,70\phantom{\,(16+ \quad \text{The quotient.}} \\
\hline
994\,025\phantom{\,(16+ \quad \text{The quotient.}} \\
994\,020\phantom{\,(16+ \quad \text{The quotient.}} \\
\hline
\end{array}$$

Finally, extracting the square root of the quotient, we have:

$$\begin{array}{l}
\dot{1}6\,(4 \text{ inches.} \quad \text{Distance of bolts from center to center.} \\
16 \\
\hline
\end{array}$$

PITCH OF BOLTS BASED ON THICKNESS OF PLATE AND STEAM PRESSURE.

RULE.—Multiply the constant whole number 28,672 *by the square of the thickness of plate, in decimals of an inch; then divide the product by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the distance of stay bolts from center to center in inches.*

Example.—Let 28,672 equal a constant.

Let 25 one hundredths of an inch equal thickness of plate.

Let 112 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{28672\times.25\times.25}{112}}=4 \text{ inches.}$$ Distance of bolts from center to center.

Performing the operation in the ordinary way, we have:

	28672	A constant.
$.25\times.25=$	.0625	Square of thickness of plate.
	14 3360	
	57 344	
	1720 32	
Dividing by the given pressure.	112) 1792.0000 (16	The quotient.
	112	
	672	
	672	

Finally, extracting the square root of the quotient, we have:

$\dot{1}6$ (4 inches. Distance of bolts from center to center.
16

THICKNESS OF PLATE BASED ON PITCH OF BOLTS AND STEAM PRESSURE.

RULE.—Multiply the given steam pressure per square inch by the square of the distance of stay bolts from center to center, in inches; then divide the product by the constant 28,672, *and extract the square root of the quotient; the answer will give the required thickness of plate in decimals of an inch.*

Example.—Let 165.67 pounds per square inch equal given steam pressure.

Let 4 inches equal distance of stay bolts from center to center.

Let 28,672 equal a constant.

Then we have:

$$\sqrt{\frac{165.67\times(4\times4)}{28672}}=.30+$$ Decimals of an inch. Thickness of plate required.

Performing the operation in the ordinary way, we have:

```
              165.67  Given steam pressure.
        4×4=      16  Square of distance between centers of bolts.
              ------
              994 02
             1656 7
             ----------
Dividing by the constant. 28672.00) 2650.72.00 (0.0924+  The quotient.
             2580 48 00
             ----------
               70 24 000
               57 34 400
               ----------
               12 89 6000
               11 46 8800
               ----------
```

Finally, extracting the square root of the quotient, we have:

```
     .0924 (.30+  Decimals of an inch. Thickness of plate required.
      9
     -----
  60) 24
```

ANOTHER RULE.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance of stay bolts from center to center, and divide the product by the constant 112, *and call the answer "The Quotient."*

Second, extract the square root of "The Quotient," and multiply the root by the constant .0625, *and the product will give the required thickness of plate in decimals of an inch.*

Example.—Let 175 pounds equal steam pressure per square inch.

Let 4 inches equal distance of stay bolts from center to center.

Let 112 equal a constant.

Let .0625 equal a constant.

Then we have:

$$\sqrt{\frac{(175\times 4^2)}{112}}\times .0625 = .3125 \text{ Decimals of an inch. Thickness of plate.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl} & 175 & \text{Steam pressure in pounds.} \\ 4\times 4= & 16 & \text{Square of distance between centers of bolts.} \\ \hline & 1050 & \\ & 175 & \\ \hline \text{A constant.}\ 112) & 2800\ (25 & \text{"The Quotient."} \\ & 224 & \\ \hline & 560 & \\ & 560 & \\ \hline \end{array}$$

Next, extracting the square root of "The Quotient," we have:

$$\begin{array}{rl} \dot{2}5\ (5 & \text{Square root.} \\ 25 & \\ \hline \end{array}$$

Finally, multiplying the square root by the constant .0625, we have:

$$\begin{array}{rl} .0625 & \\ 5 & \\ \hline .3125 & \text{Decimals of an inch. Thickness of plate.} \end{array}$$

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—First, multiply the square of the diameter of the stay bolt, in decimals of an inch, by .7854, and then multiply the product by the constant whole number 6000, and call the last product "Product No. 1."

Second, obtain the number of square inches in the area stayed by the bolt by squaring the distance from center to center of the bolts, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2" (the surface to be stayed), and the quotient will give the safe-working pressure per square inch.

Example.—Let 75 one hundredths of an inch equal diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 4 inches equal distance of stay bolts from center to center.

Then we have:

$$\frac{75 \times .75 \times .7854 \times 6000}{4 \times 4} = 165.67+ \text{ lbs. Safe-working pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

```
          .75   Diameter of bolt.
          .75   Diameter of bolt.
          ---
          375
         525
         ----
        .5625   Square of diameter of bolt.
        .7854   A constant.
        -----
        22500
       28125
      45000
     39375
     ---------
     .44178750   Area of cross section of bolt at bottom of thread.
          6000   A constant.
     ---------
 2650.72500000   "Product No. 1"
```

Next we have:

```
 4   Distance between centers of bolts.
 4   Distance between centers of bolts.
 --
 16  "Product No. 2." Number of square inches in the area of surface stayed by bolt.
```

Then, dividing "Product No. 1" by "Product No. 2," the area of surface stayed by bolt, we have:

```
16) 2650.725 (165.67+ lbs. Safe-working pressure per square inch.
    16
    ---
    105
     96
    ---
      90
      80
     ---
      107
       96
      ---
      112
      112
      ---
        5
```

STEAM PRESSURE BASED ON THICKNESS OF PLATE AND PITCH OF BOLTS.

RULE.—Multiply the constant whole number 28,672 *by the square of the thickness of plate, in decimals of an inch, and divide the product by the square of the distance, in inches, of the stay bolts from center to center; the quotient will give the safe-working pressure in pounds.*

Example.—Let 28,672 equal a constant.

Let 25 one hundredths of an inch equal thickness of plate.

Let 4 inches equal pitch of stay bolts from center to center.

Then we have:

$$\frac{28672 \times .25 \times .25}{4 \times 4} = 112 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl}
 & 28672 & \text{A constant.} \\
.25 \times .25 = & .0625 & \text{Square of thickness of plate.} \\
\hline
 & 14\,3360 & \\
 & 57\,344 & \\
 & 1720\,32 & \\
\hline
\end{array}$$

Dividing by square of distance between centers of bolts.

$$\begin{array}{rl}
4 \times 4 = 16) & 1792.0000 \text{ (112 lbs. Safe-working pressure.} \\
 & 16 \\
\hline
 & 19 \\
 & 16 \\
\hline
 & 32 \\
 & 32 \\
\hline
\end{array}$$

STEAM PRESSURE—UNITED STATES GOVERNMENT RULE.

The United States government rule is as follows for material not over seven-sixteenths inch thick:

RULE.—Multiply the constant whole number 112 *by the square of the thickness of plate, in sixteenths of an inch, and divide the product by the square of the distance from center to center of stay bolts, in inches; the quotient will give the safe-working pressure.*

Example.—Let 112 equal a constant.

Let $\frac{1}{4}$ of an inch equal thickness of plate.

Let 4 inches equal distance of stay bolts from center to center.

Then we have:

$$\tfrac{1}{4} = \tfrac{4}{16}$$

Next we have:

$$\frac{112 \times 4 \times 4}{4 \times 4} = 112 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

	112	A constant.
$4\times4=$	16	Square of thickness of plate.
	672	
	112	
Dividing by square of distance between centers of bolts. $4\times4=$ 16)	1792 (112 lbs.	Safe-working pressure.
	16	
	19	
	16	
	32	
	32	

This rule looks very simple until we consider the fact that boiler plate is seldom made in sixteenths of an inch, but nearly always in decimals of an inch; it then becomes apparent that the rules which precede the above are much simpler and much easier comprehended.

CONSTRUCTION OF FLAT SURFACES AND STEAM PRESSURE ALLOWABLE. PLATES OVER 7-16 OF AN INCH THICK.

STRAIN ON BOLTS LIMITED TO 6000 POUNDS PER SQUARE INCH OF SECTION.

Flat surfaces of furnaces, fire boxes and back connections; screw stay bolts and nuts, or plain bolts with single nut and socket, or riveted head and socket, or screw stay bolts, ends riveted, pitch limited to $10\frac{1}{2}$ inches; screw stay bolts, without socket, not allowed where salt water is used in boilers.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 6000, and call the answer "The Quotient."

Second, divide "The Quotient" by .7854, and extract the square root of the last quotient; the answer will give the required diameter of stay bolt at the bottom of thread.

Example.—Let 175 pounds equal given steam pressure per square inch.

Let 8 inches equal pitch of stay bolts from center to center.

Let 6000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{175\times8^2}{6000}\right)\div.7854}=1.54+\text{ inches.}$$ Diameter of bolt at bottom of thread.

Performing the operation in the ordinary way, we have:

```
                                   175  Steam pressure per square inch.
                         8×8=       64  Square of distance between centers of bolts.
                                   ———
                                   700
                                  1050
                                  ————
Dividing by the constant.  6000) 11200 (1.8666+   "The Quotient."
                                  6000
                                  —————
                                  52000
                                  48000
                                  —————
                                   40000
                                   36000
                                   —————
                                    40000
                                    36000
                                    —————
                                     40000
                                     36000
                                     —————
```

Next, dividing "The Quotient" by .7854, we have:

```
7854) 1.8666 (2.3766+   The last quotient.
      1 5708
      ——————
       29580
       23562
       —————
        60180
        54978
        —————
         52020
         47124
         —————
          48960
          47124
          —————
```

Finally, extracting the square root of the last quotient, we have:

```
       ·  ·  ·
       2.3766 (1.54+ inches.   Diameter of bolt at bottom
       1                                of thread.
       —
 25)   1 37
       1 25
       ————
304)   1266
       1216
       ————
```

PITCH OF BOLTS BASED ON DIAMETER OF BOLTS AND STEAM PRESSURE.

RULE.—First, multiply the area of cross section of the stay bolt at the bottom of thread by 6000, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the required distance, in inches, from center to center of stay bolts.*

Example.—Let 1.54 inches equal diameter of the bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant,

Let 175 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{1.54^2 \times .7854 \times 6000}{175}} = 8\text{— inches.}$$ Distance of bolts from center to center.

Performing the operation, we have:

```
         1.54   Diameter of bolt.
         1.54   Diameter of bolt.
         ----
          616
         770
        1 54
       ------
       2.3716   Square of diameter of bolt.
        .7854   A constant.
       ------
        94864
      118580
     189728
    1 66012
   ----------
   1.86265464   Area of cross section of bolt at
                bottom of thread.
         6000   A constant.
 --------------
 11175.92784000 "Product No. 1."
```

Next, dividing "Product No. 1" by the given steam pressure (175 pounds per square inch), we have:

```
175) 11175.9278 (63.8624+   The quotient.
     1050
     ----
      675
      525
     ----
      1509
      1400
      ----
       1092
       1050
       ----
         427
         350
         ---
          778
          700
          ---
```

Finally, extracting the square root of the quotient, we have:

```
        63.8624 (7.99+ inches.   Distance of bolts from center
        49                       to center.
        ----
  149)  1486
        1341
        ----
 1589)  14524
        14301
        -----
```

PITCH OF BOLTS BASED ON THICKNESS OF PLATE AND STEAM PRESSURE.

RULE.—Multiply the constant whole number 30,720 *by the square of the thickness of plate, in decimals of an inch; then divide the product by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the distance of stay bolts from center to center in inches.*

Example.—Let 30,720 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 156.73 pounds per square inch equal given pressure.

Then we have:

$$\sqrt{\frac{30720\times.75\times.75}{156.73}}=10.5 \text{ inches.}$$ Distance of bolts from center to center.

Performing the operation in the ordinary way, we have:

```
                   30720   A constant.
      .75×.75=     .5625   Square of thickness of plate.
                  ------
                 15 3600
                 61 440
               1843 20
              15360 0
              ------------
Dividing by the given  156.73) 17280.0000 (110.25+   The quotient.
pressure.                      15673
                               -----
                                16070
                                15673
                                -----
                                  39700
                                  31346
                                  -----
                                   83540
                                   78365
                                   -----
```

Next, extracting the square root of the quotient, we have:

```
      . . .
      110.25 (10.5 inches.   Distance of bolts from center to center.
      1
     ------
205) 1025
     1025
     ----
```

THICKNESS OF PLATE BASED ON PITCH OF BOLTS AND STEAM PRESSURE.

RULE.—Multiply the given steam pressure per square inch by the square of the distance of stay bolts from center to center, in inches; then divide the product by the constant 30,720, *and extract the square root of the quotient; the answer will give the required thickness of plate in decimals of an inch.*

Example.—Let 156.72 pounds per square inch equal given steam pressure.

Let 10.5 inches equal distance of stay bolts from center to center.

Let 30,720 equal a constant.

Then we have: $$\sqrt{\frac{156.72\times10.5^2}{30720}}=.75\text{—}$$ Decimals of an inch. Thickness of plate required.

Performing the operation in the ordinary way, we have:

```
                          156.72   Given steam pressure.
          10.5×10.5=      110.25   Square of distance between centers of bolts.
                         -------
                          7 8360
                         31 344
                       1567 2
                      15672
                     ----------
Dividing by the  30720.0000) 17278.3800.0 (0.5624+   The quotient.
constant.                    15360 0000 0
                             -------------
                              1918 3800 00
                              1843 2000 00
                              -------------
                                75 1800 000
                                61 4400 000
                                ------------
                                13 7400 0000
                                12 2880 0000
                                ------------
```

Next, extracting the square root of the quotient, we have

```
       . .
     .5624 (.75—   Decimals of an inch. Thickness of plate required.
      49
     -----
145)  724
      725
     -----
```

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—Multiply the area of cross section of the bolt at the bottom of thread by 6000; *then divide the product by the square of the distance from center to center of stay bolts, and the quotient will give the steam pressure per square inch allowable.*

Example.—Let 1.75 inches equal diameter of bolt at bottom of thread.
Let .7854 equal a constant.
Let 6000 equal a constant.
Let 10.375 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{1.75^2 \times .7854 \times 6000}{10.375 \times 10.375} = 134+ \text{ lbs. Steam pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

1.75	Diameter of bolt.
1.75	Diameter of bolt.
875	
1 225	
1 75	
3.0625	Square of diameter of bolt.
.7854	A constant.
122500	
153125	
245000	
2 14375	
2.40528750	Area of cross section of bolt at bottom of thread.
6000	A constant.

10.375 × 10.375 = 107.640625) 14431.72500000 (134+ lbs. Steam pressure per square inch.

10764 0625
3667 66250
3229 21875
438 443750
430 562500

STEAM PRESSURE BASED ON THICKNESS OF PLATE AND PITCH OF BOLTS.

RULE.—Multiply the constant whole number 30,720 *by the square of the thickness of plate, in decimals of an inch, and divide the product by the square of the distance, in inches, of the stay bolts from center to center; the quotient will give the safe-working pressure in pounds.*

Example.—Let 30,720 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 10.5 inches equal pitch of stay bolts from center to center.

Then we have:

$$\frac{30720 \times .75 \times .75}{10.5 \times 10.5} = 156.73+ \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

```
                                                30720   A constant.
                                 .75×.75=       .5625   Square of thickness of plate.
                                              -------
                                              15 3600
                                              61 440
                                            1843 20
                                           15360 0
                                           ----------
Dividing by square
of distance between  10.5×10.5=110.25) 17280.0000 (156.73+lbs. Safe-working pressure.
centers of bolts.                      11025
                                       -----
                                        62550
                                        55125
                                        -----
                                         74250
                                         66150
                                         -----
                                          81000
                                          77175
                                          -----
                                           38250
                                           33075
                                           -----
```

CONSTRUCTION OF FLAT SURFACES AND STEAM PRESSURE ALLOWABLE.
PLATES OVER 7-16 OF AN INCH THICK.

STRAIN ON BOLTS LIMITED TO 6000 POUNDS PER SQUARE INCH OF SECTION.

Flat surfaces other than furnaces, fire boxes and back connections; stay bolts, with ends threaded, having nuts on both inside and outside of plate.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 6000, and call the answer "The Quotient."

Second, divide "The Quotient" by .7854, and extract the square root of the last quotient; the answer will give the required diameter of stay bolt at the bottom of thread.

Example.—Let 175 pounds equal given steam pressure per square inch.

Let 14 inches equal pitch of stay bolts from center to center.

Let 6000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{175\times 14^2}{6000}\right)\div .7854}=2.69+ \text{ inches.}$$ **Diameter of bolt at bottom of thread.**

Performing the operation in the ordinary way, we have:

```
              175    Steam pressure per square inch.
   14×14 =    196    Square of distance between centers of bolts.
             ----
             1050
            1575
            175
            -----
Dividing by the constant.  6000) 34300 (5.7166+   "The Quotient."
                                 30000
                                 -----
                                  43000
                                  42000
                                  -----
                                   10000
                                    6000
                                   -----
                                    40000
                                    36000
                                    -----
                                     40000
                                     36000
                                     -----
```

Next, dividing "The Quotient" by .7854, we have:

```
.7854) 5.7166 (7.2785+   The last quotient.
       5 4978
       ------
        21880
        15708
        -----
         61720
         54978
         -----
          67420
          62832
          -----
           45880
           39270
           -----
```

Finally, extracting the square root of the last quotient, we have:

```
        7.2785 (2.69+ inches.   Diameter of bolt at bottom of thread.
        4
        ----
   46)  327
        276
        ----
  529)  5185
        4761
        ----
```

PITCH OF BOLTS BASED ON DIAMETER OF BOLTS AND STEAM PRESSURE.

RULE.—First, multiply the area of cross section of the stay bolt at the bottom of thread by 6000, and call the last product "Product No. 1."

Second, divide "Product No. 1" by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the required distance, in inches, from center to center of stay bolts.

Example.—Let 2.07 inches equal diameter of the bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 140 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{2.07^2 \times .7854 \times 6000}{140}} = 12 \text{ inches.} \quad \text{Required pitch of bolts.}$$

Performing the operation in the ordinary way, we have:

```
        2.07   Diameter of bolt.
        2.07   Diameter of bolt.
      ------
        1449
       4 14
      ------
      4.2849   Square of diameter of bolt.
       .7854   A constant.
      ------
      171396
     214245
    342792
   2 99943
  ----------
  3.36536046   Area of cross section of bolt at bottom of thread.
        6000   A constant.
  --------------
20192.16276000  "Product No. 1."
```

Next, dividing "Product No. 1" by the given steam pressure (140 pounds per square inch), we have:

$$
\begin{array}{rl}
140)\ 20192.16276\ (144+ & \text{The quotient.} \\
140 & \\
\hline
619 & \\
560 & \\
\hline
592 & \\
560 & \\
\hline
\end{array}
$$

Finally, extracting the square root of the quotient, we have:

$$
\begin{array}{rl}
\dot{1}4\dot{4}\ (12 \text{ inches.} & \text{Distance of bolts from center to center.} \\
1 & \\
\hline
22)\ 44 & \\
44 & \\
\hline
\end{array}
$$

PITCH OF BOLTS BASED ON THICKNESS OF PLATE AND STEAM PRESSURE.

RULE.—Multiply the constant whole number 35,840 by the square of the thickness of plate, in decimals of an inch; then divide the product by the given steam pressure per square inch, and then extract the square root of the quotient; the answer will give the distance of stay bolts from center to center in inches.

Example.—Let 35,840 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 140 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{35840\times.75\times.75}{140}}=12 \text{ inches.} \quad \text{Distance of bolts from center to center.}$$

Performing the operation in the ordinary way, we have:

$$
\begin{array}{rrl}
 & 35840 & \text{A constant.} \\
.75\times.75= & .5625 & \text{Square of thickness of plate.} \\
\hline
 & 17\ 9200 & \\
 & 71\ 680 & \\
 & 2150\ 40 & \\
 & 17920\ 0 & \\
\hline
\text{Dividing by the given steam pressure.}\quad 140) & 20160.0000\ (144 & \text{The quotient.} \\
 & 140 & \\
\hline
 & 616 & \\
 & 560 & \\
\hline
 & 560 & \\
 & 560 & \\
\hline
\end{array}
$$

Next, extracting the square root of the quotient, we have:

$$\begin{array}{r l}
 & \dot{1}4\dot{4}\,(12 \text{ inches.} \\
 & 1 \\
\hline
22) & 44 \\
 & 44 \\
\hline
\end{array}$$

Distance of bolts from center to center

THICKNESS OF PLATE BASED ON PITCH OF BOLTS AND STEAM PRESSURE.

RULE.—Multiply the given steam pressure per square inch by the square of the distance of stay bolts from center to center, in inches; then divide the product by the constant 35,840, *and extract the square root of the quotient; the answer will give the required thickness of plate in decimals of an inch.*

Example.—Let 140 pounds per square inch equal given steam pressure.

Let 12 inches equal distance of stay bolts from center to center.

Let 35,840 equal a constant.

Then we have:

$$\sqrt{\frac{140\times12\times12}{35840}}=.75$$

Decimals of an inch. Thickness of plate required.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r r l}
 & 140 & \text{Given steam pressure.} \\
12\times12= & 144 & \text{Square of distance between centers of bolts.} \\
\hline
 & 560 & \\
 & 560 & \\
 & 140 & \\
\hline
\end{array}$$

$$\begin{array}{l r l}
\text{Dividing by the constant.} & 35840\,)\,20160.0\,(\,0.5625 & \text{The quotient.} \\
 & 17920\,0 & \\
\hline
 & 2240\,00 & \\
 & 2150\,40 & \\
\hline
 & 89\,600 & \\
 & 71\,680 & \\
\hline
 & 17\,9200 & \\
 & 17\,9200 & \\
\hline
\end{array}$$

Next, extracting the square root of the quotient, we have

$$\begin{array}{r l}
 & .\dot{5}6\dot{2}5\,(.75 \\
 & 49 \\
\hline
145) & 725 \\
 & 725 \\
\hline
\end{array}$$

Decimals of an inch. Thickness of plate required.

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—Multiply the area of cross section of the bolt at the bottom of thread by 6000; *then divide the product by the square of the distance from center to center of stay bolts, and the quotient will give the steam pressure per square inch allowable.*

Example.—Let 2.07 inches equal diameter of bolt at bottom of thread.
Let .7854 equal a constant.
Let 6000 equal a constant.
Let 14 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{2.07^2 \times .7854 \times 6000}{14 \times 14} = 103+ \text{ lbs. Steam pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl}
2.07 & \text{Diameter of bolt.} \\
2.07 & \text{Diameter of bolt.} \\
\hline
1449 & \\
4\,14 & \\
\hline
4.2849 & \text{Square of diameter of bolt.} \\
.7854 & \text{A constant.} \\
\hline
171396 & \\
214245 & \\
342792 & \\
2\,99943 & \\
\hline
3.36536046 & \text{Area of cross section of bolt at bottom of thread.} \\
6000 & \text{A constant.} \\
\hline
\end{array}$$

Dividing by square of distance between centers of bolts. $14 \times 14 = 196)\ 20192.16276000\ (103+$ lbs. Steam pressure per square inch.

$$\begin{array}{r}
196 \\
\hline
592 \\
588 \\
\hline
\end{array}$$

STEAM PRESSURE BASED ON THICKNESS OF PLATE AND PITCH OF BOLTS.

RULE.—Multiply the constant whole number 35,840 *by the square of the thickness of plate, in decimals of an inch, and divide the product by the square of the distance, in inches, of the stay bolts from center to center; the quotient will give the safe-working pressure in pounds.*

Example.—Let 35,840 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 14 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{35840 \times .75 \times .75}{14 \times 14} = 102.85+ \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

```
                                           35840  A constant.
                              .75×.75=     .5625  Square of thickness of plate.
                                         -------
                                          179200
                                          71680
                                        215040
                                       179200
                                      -----------
Dividing by square of   14×14=  196) 20160.0000 (102.85+ lbs.  Safe-working
distance between cen-                196                          pressure.
ters of bolts.                        ---
                                       560
                                       392
                                       ----
                                       1680
                                       1568
                                       ----
                                        1120
                                         980
                                        ----
```

Taking the same plate and placing the same bolts at a distance of 12 inches from center to center, and determining the safe-working pressure, we have:

$$\frac{35840 \times .75 \times .75}{12 \times 12} = 140 \text{ lbs. Safe-working pressure.}$$

Performing the operation in the ordinary way, we have:

```
                                          35840
                                          .5625
                                         -------
                                         179200
                                         71680
                                       215040
                                      179200
                                     -----------
Dividing by square of   12×12=  144) 20160.0000 (140 lbs.  Safe-working pressure.
distance between                     144
centers of bolts.                    ---
                                      576
                                      576
                                      ---
                                        0
```

CONSTRUCTION OF FLAT SURFACES AND STEAM PRESSURE ALLOWABLE. PLATES OVER 7-16 OF AN INCH THICK.

STRAIN ON BOLTS LIMITED TO 6000 POUNDS PER SQUARE INCH OF SECTION.

Flat surfaces other than furnaces, fire boxes and back connections; stay bolts with double nuts, and a washer at least one-half the thickness of plate, and in size equal to two-fifths of pitch of stay bolts, or plates fitted with double angle iron, riveted to plate with leaf at least two-thirds of the thickness of plate, and depth at least one-fourth of the pitch.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 6000, *and call the answer "The Quotient."*

Second, divide "The Quotient" by .7854, *and extract the square root of the last quotient; the answer will give the required diameter of stay bolt at the bottom of thread.*

Example.—Let 175 pounds equal given steam pressure per square inch.

Let 15 inches equal pitch of stay bolts from center to center.

Let 6000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{175\times15^2}{6000}\right)\div.7854}=2.88+\text{ inches.}$$ Diameter of bolt at bottom of thread.

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl} & 175 & \text{Steam pressure per square inch.} \\ 15\times15= & 225 & \text{Square of distance between centers of bolts.} \\ \hline & 875 & \\ & 350 & \\ & 350 & \\ \hline \end{array}$$

$$\begin{array}{rl} \text{Dividing by the constant. } 6000) & 39375\ (6.5625 \quad \text{"The Quotient."} \\ & 36000 \\ \hline & 33750 \\ & 30000 \\ \hline & 37500 \\ & 36000 \\ \hline & 15000 \\ & 12000 \\ \hline & 30000 \\ & 30000 \\ \hline \end{array}$$

Next, dividing "The Quotient" by .7854, we have:

```
.7854) 6.5625 (8.34+   The last quotient.
       6 2832
       ------
        27930
        23562
        -----
         43680
         31416
         -----
```

Finally, extracting the square root of the last quotient, we have:

```
       8.34 (2.88+inches.   Diameter of bolt at bottom of thread.
       4
       ---
  48)  434
       384
       ----
 568)  5000.
       4544
       ----
```

PITCH OF BOLTS BASED ON DIAMETER OF BOLTS AND STEAM PRESSURE.

RULE.—First, multiply the area of cross section of the stay bolt at the bottom of thread by 6000, *and call the last product "Product No.* 1."

Second, divide "Product No. 1" *by the given steam pressure per square inch; then extract the square root of the quotient, and the answer will give the required distance, in inches, from center to center of stay bolts.*

Example.—Let 2.88 inches equal diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 146 pounds per square inch equal given steam pressure

Then we have:

$$\sqrt{\frac{2.88^2 \times .7854 \times 6000}{146}} = 16.36+ \text{ inches.}$$ Distance of bolts from center to center.

Performing the operation in the ordinary way, we have:

```
   2.88   Diameter of bolt.
   2.88   Diameter of bolt.
  ------
   2304
  2 304
  5 76
  ------
  8.2944  Square of diameter of bolt.
```

Am't carried forward, 8.2944

Am't brought forward, 8.2944 Square of diameter of bolt.
.7854 A constant.

```
      8.2944   Square of diameter of bolt.
       .7854   A constant.
     -------
      331776
     414720
    663552
   5 80608
   ---------
   6.51442176    Area of cross section of bolt at bottom of thread.
         6000    A constant.
  --------------
 39086.53056000  "Product No. 1."
```

Next, dividing "Product No. 1" by the given steam pressure (146 pounds per square inch), we have:

```
146) 39086.5305 (267.7159+   The quotient.
     292
     ---
      988
      876
      ----
      1126
      1022
      ----
       1045
       1022
       ----
         233
         146
         ---
          870
          730
          ----
          1405
          1314
          ----
```

Finally, extracting the square root of the quotient, we have:

```
      267.7159 (16.36+ inches.  Distance of bolts from center to center.
      1
      ---
26)   167
      156
      ----
323)  1171
       969
      -----
3266) 20259
      19596
      -----
```

PITCH OF BOLTS BASED ON THICKNESS OF PLATE AND STEAM PRESSURE.

RULE.—Multiply the constant whole number 51,200 *by the square of the thickness of plate, in decimals of an inch; then divide the product by the given steam pressure per square inch, and extract the square root of the quotient; the answer will give the distance of stay bolts, in inches, from center to center.*

Example.—Let 51,200 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 146 pounds per square inch equal given steam pressure.

Then we have:

$$\sqrt{\frac{51200\times.75^2}{146}}=14+\text{ inches. Distance of bolts from center to center.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl}
 & 51200 & \text{A constant.}\\
.75\times.75= & .5625 & \text{Square of thickness of plate.}\\
\hline
 & 25\,6000 & \\
 & 102\,400 & \\
 & 3072\,00 & \\
 & 25600\,0 & \\
\hline
\text{Dividing by the given pressure.}\ 146) & 28800.0000(\,197+ & \text{The quotient.}\\
 & 146 & \\
\hline
 & 1420 & \\
 & 1314 & \\
\hline
 & 1060 & \\
 & 1022 & \\
\hline
\end{array}$$

Finally, extracting the square root of the quotient, we have:

$$\begin{array}{rl}
 & \dot{1}9\dot{7}(14+\text{ inches. Distance of bolts from center to center.}\\
 & 1\\
\hline
24) & 97\\
 & 96\\
\hline
\end{array}$$

THICKNESS OF PLATE BASED ON PITCH OF BOLTS AND STEAM PRESSURE

RULE.—Multiply the given steam pressure per square inch by the square of the distance of stay bolts from center to center, in inches; then divide the product by the constant 51,200, *and extract the square root of the quotient; the answer will give the required thickness of plate in decimals of an inch.*

Example.—Let 128 pounds per square inch equal given steam pressure.

Let 15 inches equal distance of stay bolts from center to center.

Let 51,200 equal a constant.

Then we have:

$$\sqrt{\frac{128\times15\times15}{51200}}=.75$$ Decimals of an inch. Thickness of plate required.

Performing the operation in the ordinary way, we have:

```
              128   Given steam pressure.
     15×15=   225   Square of distance between centers of bolts.
              ---
              640
             256
            256
           ------
Dividing by the constant.   51200) 28800.0 (0.5625   The quotient.
                                   25600 0
                                   -------
                                    3200 00
                                    3072 00
                                    -------
                                     128 000
                                     102 400
                                     -------
                                      25 6000
                                      25 6000
                                      -------
```

Next, extracting the square root of the quotient, we have:

```
         .5625 (.75   Decimals of an inch.  Thickness of plate required.
          49
         ----
   145)  725
         725
         ---
```

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—Multiply the area of cross section of the stay bolt at the bottom of thread by 6000; then divide the last product by the square of the distance, in inches, from center to center of the stay bolts; the quotient will give the steam pressure in pounds per square inch.

Example.—Let 2.88 inches equal diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 15 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{2.88^2\times.7854\times6000}{15\times15}=173.71+\text{ lbs.}$$ Steam pressure per square inch.

Performing the operation in the ordinary way, we have:

```
                         2.88   Diameter of bolt.
                         2.88   Diameter of bolt.
                       -------
                         2304
                       2 304
                       5 76
                       -------
                       8.2944   Square of diameter of bolt.
                        .7854   A constant.
                       -------
                       331776
                      414720
                     663552
                   5 80608
                   ----------
                   6.51442176      Area of cross section of bolt at bottom of thread.
                         6000   A constant.
                   ---------------
Dividing by square of distance of bolts between centers.
15×15= 225) 39086.53056000 (173.71+ lbs.  Steam pressure per square inch.
            225
            ----
            1658
            1575
            ----
              836
              675
              ----
              1615
              1575
              ----
                403
                225
                ---
```

STEAM PRESSURE BASED ON THICKNESS OF PLATE AND PITCH OF BOLTS.

RULE.—Multiply the constant whole number 51,200 *by the square of the thickness of plate, in decimals of an inch, and divide the product by the square of the distance, in inches, of the stay bolts from center to center; the quotient will give the steam pressure per square inch allowable.*

Example.—Let 51,200 equal a constant.

Let 75 one hundredths of an inch equal thickness of plate.

Let 15 inches equal pitch of stay bolts from center to center.

Then we have:

$$\frac{51200 \times 75^2}{15 \times 15} = 128 \text{ lbs.} \quad \text{Steam pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

$$
\begin{array}{rrl}
 & 51200 & \text{A constant} \\
.75\times.75= & .5625 & \text{Square of thickness of plate.} \\
\hline
 & 25\,6000 & \\
 & 102\,400 & \\
 & 3072\,00 & \\
 & 25600\,0 & \\
\hline
\end{array}
$$

Dividing by square of distance between centers of bolts. $15\times15=225)\,28800.0000\,(128$ lbs. Steam pressure per square inch.

$$
\begin{array}{r}
225 \\
\hline
630 \\
450 \\
\hline
1800 \\
1800 \\
\hline
\end{array}
$$

CONSTRUCTION OF FLAT SURFACES AND STEAM PRESSURE ALLOWABLE.
STEEL STAY BOLTS EXCEEDING 1 1-4 INCHES AND NOT
EXCEEDING 2 1-2 INCHES IN DIAMETER.

Steel stay bolts exceeding $1\frac{1}{4}$ inches and not exceeding $2\frac{1}{2}$ inches in diameter at bottom of thread, are allowed a strain of 8000 pounds per square inch of cross section.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 8000, and call the answer "The Quotient."

Second, divide "The Quotient" by .7854, and extract the square root of the last quotient; the answer will give the required diameter of stay bolts at the bottom of thread.

Example.—Let 175 pounds equal given steam pressure per square inch.

Let 15 inches equal pitch of bolts from center to center.

Let 8000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{175\times15^2}{8000}\right)\div.7854}=2.5+\text{ inches.}$$ Diameter of bolt at bottom of thread.

Performing the operation in the ordinary way, we have:

```
                                   175  Steam pressure per square inch.
                         15×15=    225  Square of distance between centers of bolts.
                                   ———
                                   875
                                  350
                                 350
                                 ————
Dividing by the constant. 8000) 39375 (4.9218+  "The Quotient."
                                32000
                                —————
                                 73750
                                 72000
                                 —————
                                  17500
                                  16000
                                  —————
                                   15000
                                    8000
                                   —————
                                    70000
                                    64000
                                    —————
```

Next, dividing "The Quotient" by .7854, we have:

```
.7854) 4.9218 (6.2666+  The last quotient.
       4 7124
       ——————
        20940
        15708
        —————
         52320
         47124
         —————
          51960
          47124
          —————
           48360
           47124
           —————
```

Finally, extracting the square root of the last quotient, we have:

```
     6.2666 (2.5+ inches.  Diameter of bolt at bottom of thread.
     4
     ———
45)  226
     225
     ———
```

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—Multiply the area of cross section of the stay bolt at the bottom of thread by 8000; then divide the product by the square of the distance, in inches, from center to center of stay bolts, and the quotient will give the steam pressure per square inch allowable.

Example.—Let 2.5 inches equal diameter of bolt at the bottom of thread.

Let .7854 equal a constant.

Let 8000 equal a constant.

Let 15 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{2.5^2 \times .7854 \times 8000}{15 \times 15} = 174.53+ \text{ lbs. Pressure per square inch.}$$

Performing the operation in the ordinary way, we have:

2.5	Diameter of bolt.
2.5	Diameter of bolt.
1 25	
5 0	
6.25	Square of diameter of bolt.
.7854	A constant.
2500	
3125	
5000	
4 375	
4.908750	Area of cross section of bolt at bottom of thread.
8000	A constant.

Dividing by square of distance between centers of bolts. $15 \times 15 =$ 225) 39270.000000 (174.53+ lbs. Steam pressure per square inch.

```
225) 39270.000000 (174.53+
     225
     ----
     1677
     1575
     ----
      1020
       900
      ----
       1200
       1125
       ----
         750
         675
         ---
```

STEEL STAY BOLTS EXCEEDING 2 1-2 INCHES IN DIAMETER.

Steel stay bolts exceeding a diameter of 2½ inches at the bottom of thread, are allowed a strain of 9000 pounds per square inch of cross section.

DIAMETER OF STAY BOLTS.

RULE.—First, multiply the given steam pressure per square inch by the square of the distance between centers of stay bolts, and divide the last product by 9000, and call the answer "The Quotient."

Second, divide "The Quotient" by .7854, and extract the square root of the last quotient; the answer will give the required diameter of stay bolt at the bottom of thread.

Example.—Let 208.81 pounds equal given steam pressure per square inch.

Let 16 inches equal pitch of bolts from center to center.

Let 9000 equal a constant.

Let .7854 equal a constant.

Then we have:

$$\sqrt{\left(\frac{208.81\times16^2}{9000}\right)\div.7854}=2.74+\text{ inches.}$$ **Diameter of bolt at bottom of thread.**

Performing the operation in the ordinary way, we have:

```
                 208.81   Steam pressure per square inch.
      16×16=        256   Square of distance between centers of bolts.
                -------
                1252 86
               10440 5
               41762
               --------
Dividing by the constant. 9000.00) 53455.36 (5.9394+   "The Quotient."
                                   45000 00
                                   --------
                                    8455 360
                                    8100 000
                                    --------
                                     355 3600
                                     270 0000
                                     --------
                                      85 36000
                                      81 00000
                                      --------
                                       4 360000
                                       3 600000
```

Next, dividing "The Quotient" by .7854, we have:

```
.7854) 5.9394 (7.5622+   The last quotient.
       5 4978
       ------
        44160
        39270
        -----
         48900
         47124
         -----
          17760
          15708
          -----
           20520
           15708
           -----
```

Finally, extracting the square root of the last quotient, we have:

```
       7.5622 (2.75— inches. Diameter of bolt at bottom
       4                                of thread.
      ----
  47) 356
      329
      ----
 545) 2722
      2725
      ----
```

STEAM PRESSURE BASED ON DIAMETER AND PITCH OF BOLTS.

RULE.—Multiply the area of cross section of the stay bolt at the bottom of thread by 9000; then divide the product by the square of the distance, in inches, from center to center of stay bolts, and the quotient will give the steam pressure per square inch allowable.

Example.—Let 2.75 inches equal diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 9000 equal a constant.

Let 16 inches equal pitch of bolts from center to center.

Then we have:

$$\frac{2.75^2 \times .7854 \times 9000}{16 \times 16} = 208.81+ \text{ lbs.}$$ Steam pressure per square inch.

Performing the operation in the ordinary way, we have:

```
        2.75      Diameter of bolt.
        2.75      Diameter of bolt.
        ----
        1375
       1 925
       5 50
       ------
       7.5625     Square of diameter of bolt.
        .7854     A constant.
       ------
       302500
      378125
     605000
    5 29375
    ----------
    5.93958750    Area of cross section of bolt at bottom
                       of thread.
          9000    A constant.
    --------------
Am't carried forward,  53456.28750000
```

Dividing by square of distance between centers of bolts. 16×16=256) 53456.28750000 (208.81+ lbs. Steam pressure per square inch.

```
16×16=256) 53456.28750000 (208.81+ lbs.
           512
           ----
            2256
            2048
            ----
             2082
             2048
             ----
               348
               256
               ---
```

FIRE PUMPS.

Section 4471 of the Revised Statutes of the United States provides that, "Every steamer exceeding 200 tons burden, and carrying passengers, shall be provided with two good double-acting fire pumps, to be worked by hand. Each chamber of such pumps, except pumps upon steamers in service on the 28th day of February, 1871, shall be of sufficient capacity to contain not less than 100 cubic inches of water."

SIZE OF PUMP REQUIRED UNDER THE PROVISIONS OF SECTION 4471 OF THE REVISED STATUTES.

RULE.—Assume any convenient and practical length of stroke of pump; then divide the number of cubic inches (100) *of water required by the length of the stroke of the pump in inches; then divide the quotient by* .7854, *and extract the square root of the last quotient; the answer will give the required diameter of the plunger.*

Example.—Let 8 inches equal assumed length of stroke of pump.
Let 100 cubic inches equal capacity of chamber of pump.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{100 \div 8}{.7854}} = 3.98+ \text{ inches.}$$

Required diameter of plunger of pump.

Performing the operation in the ordinary way, we have:

```
       8) 100
          -------
.7854) 12.5000(15.91+ square inches.
        7 854
        -----
        4 6460
        3 9270
        ------
          71900
          70686
          -----
           12140
            7854
           -----
```

Square of required diameter of plunger.

Next, extracting the square root of the quotient, we have:

```
     15.91 (3.98+ inches.  Required diameter of plunger of pump.
     9
     ----
69)  691
     621
     ----
788) 7000
     6304
     ----
```

HORSE POWER OF MARINE ENGINES.

Under the regulations of the Treasury Department, inspectors are required to enter in a book, containing particulars of inspection, the horse power of the engines of each steamer inspected by them. As the use of the indicator is not required, the actual horse power of each engine must be determined by computation, which may be done by the following rule:

RULE.—Multiply the area of the diameter of the cylinder by the average (or, in other words, the mean effective) pressure per square inch, throughout the stroke; then multiply the product by the number of feet the piston travels per minute, and divide the last product by 33,000.

Example.—Let 24 inches equal diameter of cylinder.
Let .7854 equal a constant.
Let 150 pounds equal initial pressure per square inch.
Let one-half equal point of stroke of piston at which steam is cut off.
Let 5 feet equal length of piston stroke.
Let 33,000 equal a constant.

MEAN EFFECTIVE PRESSURE.

The first thing is to determine the average (or mean effective) pressure per square inch throughout the stroke, and that is done by the aid of the following simple diagram:

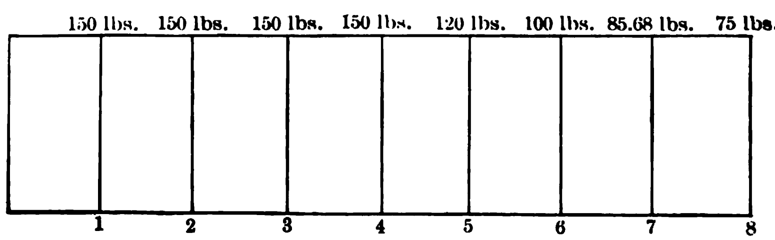


Fig. 173

(5)	(6)	(7)	(8)
5) 150	6) 150	7) 150	8) 150
30	25	21.42+	18.75
4	4	4	4
120	100	85.68	75.00

The diagram represents a steam engine cylinder. The figures at the bottom of the diagram represent different parts of the stroke of the piston, and the figures at the top represent the steam pressure at the different parts of the stroke. It will be noticed that the pressure (150 pounds) remains the same during one-half of the stroke, when it begins to fall. The steam is cut off at four-eighths, or one-half stroke, and the steam does its work expansively during the remainder of the stroke; but the pressure per square inch is decreased, as shown by the figures to the right of the point of cut off, at the top of the diagram, and these figures are obtained in the following manner:

The steam being cut off at 4, and when the piston gets to 5, we desire to determine the pressure at that point, we say $\frac{4}{5}$ of 150; at 6 we say $\frac{4}{6}$ of 150; at 7 we say $\frac{4}{7}$ of 150; at 8 we say $\frac{4}{8}$ of 150.

$\frac{4}{5}$ of 150 is:

$$\begin{array}{r} 5)\,150 \\ \hline 30 \\ 4 \\ \hline 120 \end{array}$$

120 lbs. Pressure at $\frac{5}{8}$ of the stroke.

$\frac{4}{6}$ of 150 is:

$$\begin{array}{r} 6)\,150 \\ \hline 25 \\ 4 \\ \hline 100 \end{array}$$

100 lbs. Pressure at $\frac{6}{8}$ of the stroke.

$\frac{4}{7}$ of 150 is:

$$\begin{array}{r} 7)\,150 \\ \hline 21.42+ \\ 4 \\ \hline 85.68 \end{array}$$

85.68 lbs. Pressure at $\frac{7}{8}$ of the stroke.

$\frac{4}{8}$ of 150 is:

$$\begin{array}{r} 8)\,150 \\ \hline 18.75 \\ 4 \\ \hline 75.00 \end{array}$$

75.00 lbs. Pressure at $\frac{8}{8}$ of the stroke.

It makes no difference at what part of the stroke the steam is cut off, the operation is the same. The figure at the bottom of point of cut off is always taken as the numerator, and all succeeding figures as denominators, while the pressure at the top of the point of cut off is

always taken as the dividend. So that the pressure at point of cut off is always divided by the figures at the bottom succeeding the figures at which the steam is cut off, and the quotient of each operation is multiplied by the figure at the bottom of the diagram at the point of cut off; in this way the pressure for each point in the stroke is obtained, after which all of the pressures on top of the diagram are added together, and the sum is divided by the number of divisions in the diagram, which in the present case is 8.

Thus we have:

150	1st part of stroke.
150	2nd part of stroke.
150	3rd part of stroke.
150	4th part of stroke.
120	5th part of stroke.
100	6th part of stroke.
85.68	7th part of stroke.
75	8th part of stroke.
8) 980.68	
122.585 lbs.	Average pressure throughout the stroke.

We are now prepared to determine the horse power of the engine. Proceeding according to the rule, we have:

$$\frac{24\times24\times.7854\times122.585\times2\times5\times18}{33000}=302.4887+ \text{ Horse power of engine.}$$

It will be noticed that the figure 2 is used in the example, and will, therefore, require explanation. As the piston makes two strokes during each revolution of the crank, the piston travels twice the length of one stroke, which in this case is 10 feet for each revolution of the crank, or twice 5 feet for each revolution of crank. Hence, the use of the figure 2.

Performing the operation in the ordinary way, we have:

24	Diameter of cylinder.
24	Diameter of cylinder.
96	
48	
576	Square of diameter of cylinder.
.7854	A constant.
4 7124	
54 978	
392 70	
452.3904	Area of diameter of cylinder.

Am't carried forward, 452.3904

Am't brought forward, 452.3904 Area of diameter of cylinder.
122.585 Average pressure throughout the stroke.

2 2619520
36 191232
226 19520
904 7808
9047 808
45239 04

55456.2771840
2 Number of strokes of piston to one revolution of crank.

110912.5543680
5 Length of stroke.

554562.7718400
18 Number of revolutions of crank.

4436502 1747200
5545627 718400

33000) 9982129.8931200 (302.48+ Horse power of engine.
99000

82129
66000

161298
132000

292989
264000

SAFETY-VALVES.

[Sections 4418, 4419, 4436 and 4437 of the Revised Statutes of the United States. Section 24, of Rule 2; Section 5, of Rule 5 and Section 14, of Rule 9, of the United States Board of Supervising Inspectors.]

Section 4418 of the Revised Statutes requires inspectors to "satisfy themselves that the safety-valves are of suitable dimensions, sufficient in number and well arranged; and that the weights of the safety-valves are properly adjusted, so as to allow no greater pressure in the boilers than the amount prescribed in the certificate of inspection." Section 4419 gives the inspectors authority to take one of the safety-valves "wholly from the control of all persons engaged in navigating a steam vessel" if they find it necessary to do so. Section 4436 provides that "every boiler shall be provided with a good, well-constructed safety-valve or valves, of such number, dimensions and arrangements as shall be prescribed by the Board of Supervising Inspectors." Section 4437 reads as follows:

"SEC. 4437. Every person who intentionally loads or obstructs, or causes to be loaded or obstructed, in any manner, the safety-valve of a boiler, or who employs any other means or device whereby the boiler may be subjected to a greater pressure than the amount allowed by the certificate of the inspectors, or who intentionally deranges or hinders the operation of any machinery or device employed to denote the state of the water or steam in any boiler, or to give warning of approaching danger, or who intentionally permits the water to fall below the prescribed low water line of the boiler, and every person concerned therein, directly or indirectly, shall be guilty of a misdemeanor, and shall be fined two hundred dollars, and may also be imprisoned not exceeding five years."

Section 24, of Rule 2, of the Board of Supervising Inspectors, relates to the construction of lever safety-valves, and to the required capacity of lever and spring-loaded valves.

Section 5, of Rule 5, provides that "no person shall receive license (as an engineer) who is not able to determine the weight necessary to be placed on the lever of a safety-valve (the diameter of the valve, length of lever and distance of valve from fulcrum being given) to withstand any given pressure of steam in a boiler, or who is not able to figure and determine the strain brought on the braces of a boiler with a given pressure of steam, the position and distance apart of braces being known; such knowledge to be determined by an examination in writing, and the report of examination filed with the application in the office of the local inspectors, and no engineer, or assistant engineer, now holding a license, shall have the grade of the same raised without possessing the above qualifications."

Section 14, of Rule 9, "in case it comes to the knowledge of inspectors that steam has been carried in any boiler in excess of that allowed by law, such inspectors are advised to report the facts to the United States District Attorney for prosecution under Section 4437 of the Revised Statutes, and in addition thereto require the owners to place on the boilers a lock-up safety-valve that will prevent the carrying of any excess of steam, and such valve shall be exclusively under the control of the inspectors."

LEVER SAFETY-VALVES.

Section 24, of Rule 2, of the Board of Supervising Inspectors, provides, among other things, the following:

"Lever safety-valves to be attached to marine boilers shall have an area of not less than one square inch to two square feet of the grate surface in the boiler, and the seats of all such safety-valves shall have an angle of inclination of 45 degrees to the center line of their axis."

AREA OF A LEVER SAFETY-VALVE.

RULE.—Divide the number of square feet of the grate surface of the boiler by 2, and the quotient will give the required area of a lever safety-valve in square inches.

Example.—Let 4 feet equal width of grate surface.
Let 5 feet equal length of grate surface.
Let 2 equal a divisor.

Then we have:

$$\frac{4\times5}{2}=10 \text{ square inches.}$$ Required area of lever safety valve.

Performing the operation in the ordinary way, we have:

4 Width of grate surface.
5 Length of grate surface.

Dividing by the divisor. 2) 20

10 square inches. Required area of lever safety-valve.

DIAMETER OF LEVER SAFETY-VALVE.

RULE.—First, multiply the length of the grate surface, in feet, by the width of the grate surface, in feet, and divide the product by 2; the quotient will give the required area of the valve in square inches.

Second, divide the required area of the valve by .7854, and extract the square root of the quotient; the answer will give the required diameter of the valve in inches.

Example.—Let 5 feet equal length of grate surface.
Let 4 feet equal width of grate surface.
Let 2 equal the divisor.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{(5\times4)\div2}{.7854}}=3.56+ \text{ inches.}$$ Required diameter of valve.

Performing the operation in the ordinary way, we have:

5 Length of grate surface.
4 Width of grate surface.

2) 20

Am't carried forward, 10

Am't brought forward, .7854) 10.0000 (12.7323+ **The quotient.**

```
7854
-----
21460
15708
-----
 57520
 54978
 -----
  25420
  23562
  -----
   18580
   15708
   -----
    28720
    23562
    -----
```

Next, extracting the square root of the quotient, we have:

12.7323 (3.56+ inches. **Required diameter of valve.**

```
      12.7323 (3.56+
      9
      ---
 65)  373
      325
      ---
706)  4823
      4236
      ----
```

SPRING-LOADED SAFETY-VALVES.

The third paragraph of Section 24, of Rule 2, reads as follows:

"Any spring-loaded safety-valve constructed so as to give an increased lift by the operation of steam, after being raised from its seat, or any spring-loaded safety-valve constructed in any other manner so as to give an effective area of opening equal to that of the aforementioned spring-loaded safety-valve may be used in lieu of the common lever weighted valve on all boilers on steam vessels, and all such spring-loaded safety-valves shall be required to have an area of not less than one square inch to three square feet of the grate surface of the boiler; and each spring-loaded safety-valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth of the diameter of the valve opening; and the seats of all such safety-valves shall have an angle of inclination to the center line of their axis of 45 degrees. But in no case shall any spring-loaded safety-valve be used in lieu of the lever safety-valve without the approval of the Board of Supervising Inspectors."

AREA OF SPRING-LOADED SAFETY-VALVES.

RULE.—Divide the number of square feet of the grate surface of the boiler by 3, and the quotient will give the area of the valve in square inches.

Example.—Let 5 feet equal length of grate surface.
Let 4 feet equal width of grate surface.
Let 3 equal a divisor.

Then we have:

$$\frac{5\times4}{3}=6.6666+ \text{ square inches. Required area of valve.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 5 \\ 4 \\ \hline 3)\,20.0000 \\ \hline 6.6666+ \end{array} \text{ square inches. Required area of valve.}$$

DIAMETER OF SPRING-LOADED SAFETY-VALVES.

RULE.—First, multiply the length of the grate surface, in feet, by the width of the grate surface, in feet, and divide the product by 3; *the quotient will give the required area of the valve in square inches.*

Second, divide the required area of the valve by .7854, *and extract the square root of the quotient; the answer will give the required diameter of the valve in inches.*

Example.—Let 5 feet equal length of grate surface.
Let 4 feet equal width of grate surface.
Let 3 equal the divisor.
Let .7854 equal a constant.

Then we have:

$$\sqrt{\frac{(5\times4)\div3}{.7854}}=2.91+ \text{ inches. Required diameter of valve.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{l} 5 \\ 4 \\ \hline 3)\ 20.0000 \\ \hline .7854)\ 6.6666\ (8.4881+ \text{ The quotient.} \\ 6\ 2832 \\ \hline 38340 \\ 31416 \\ \hline 69240 \\ 62832 \\ \hline 64080 \\ 62832 \\ \hline 12480 \\ 7854 \\ \hline \end{array}$$

Next, extracting the square root of the quotient, we have:

```
      8.4881 (2.91+ inches.  Required diameter of valve.
      4
     ----
 49) 448
     441
     --------
  581) 781
       581
       ---
```

PRACTICAL ADJUSTMENT OF SAFETY-VALVES.

As the adjustment of safety-valves, mathematically, has been exhaustively treated in a preceding chapter, it will not be necessary to go extensively into the subject here; hence, the instructions will be confined to the illustration of the more complicated questions connected with safety-valves, met in practice by inspectors and engineers. In illustrating this subject, the valves heretofore employed were small and made simple for the purpose of enabling the student to more easily understand the rules and examples given. The rules laid down here will be exactly the same as those given in the fore part of this work, but they will now be applied to the safety-valves in general use; therefore, the information that will now be imparted will be strictly practical, while that previously given was merely theoretical. The student will therefore have an opportunity to make a practical application of the information given in the theory of the adjustment of safety-valves mathematically.

PRESSURE PER SQUARE INCH REQUIRED TO RAISE A VALVE.

[Section 4418 of the Revised Statutes of the United States.]

RULE.—First, multiply the weight of the valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 1."

Second, multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the weight by the distance of its centers from the fulcrum, and call the product "Product No. 3."

Fourth, add Products Nos. 1, 2 and 3 together, and call the answer "The Sum."

Fifth, square the diameter of the valve and multiply the square of its diameter by .7854, and multiply the product by the distance of the center of the valve from the fulcrum, and call the last product "Product No. 4."

Sixth, divide "The Sum" by "Product No. 4," and the quotient will give the pressure per square inch required to raise the valve.

Example.—Let 4 inches equal distance of center of valve from fulcrum.

Let 9 pounds equal weight of valve and spindle.

Let 20 inches equal distance of center of gravity of lever from fulcrum.

Let 35 pounds equal weight of lever.

Let 32 inches equal a distance of weight on lever from fulcrum.

Let 180 pounds equal weight of the weight.

Let .7854 equal a constant.

Let 4 inches equal diameter of the valve.

The valve and attachments being arranged as shown in Fig. 174.

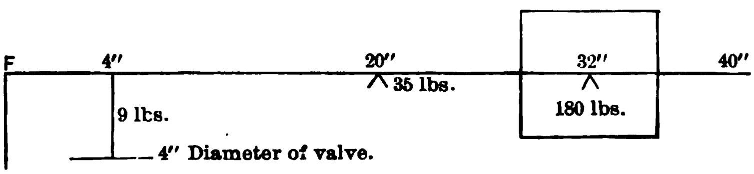


Fig. 174

Then we have:

$$\frac{(9\times4)+(35\times20)+(180\times32)}{4\times4\times.7854\times4}=129.23+\text{ lbs.}$$ Pressure per square inch required to raise the valve.

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl} 9\times4= & 36 & \text{“Product No. 1.”} \\ 35\times20= & 700 & \text{“Product No. 2.”} \\ 180\times32= & 5760 & \text{“Product No. 3.”} \\ \hline & 6496 & \text{“The Sum.”} \end{array}$$

Next we have:

$$\begin{array}{rl} 4 & \text{Diameter of valve.} \\ 4 & \text{Diameter of valve.} \\ \hline 16 & \text{Square of the diameter of valve.} \\ .7854 & \text{A constant.} \\ \hline 4\,7124 & \\ 7\,854 & \\ \hline 12.5664 & \text{Area of valve.} \\ 4 & \text{Distance of center of valve from the fulcrum.} \\ \hline 50.2656 & \text{“Product No. 4.”} \end{array}$$

Finally, dividing "The Sum" (6496) by "Product No. 4" (50.2656), we have:

```
50.2656) 6496.0000 (129.23+ lbs.   Pressure required to raise
         5026 56                     the valve.
         -------
         1469 440
         1005 312
         --------
          464 1280
          452 3904
          ---------
           11 73760
           10 05312
           --------
            1 684480
            1 507968
            --------
```

DISTANCE FROM FULCRUM A GIVEN WEIGHT IS REQUIRED TO BE PLACED ON THE LEVER TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE.

[Section 4418 of the Revised Statutes of the United States, and Section 5, of Rule 5, of the United States Board of Supervising Inspectors.]

RULE.—First, multiply the area of the valve by the distance of the center of the valve from the fulcrum; then multiply the product by the given pressure, in pounds per square inch, and call the product "Product No. 1."

Second, multiply the weight of the valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and call the product "Product No. 3."

Fourth, add Products Nos. 2 and 3 together, and call the answer "The Sum;" then subtract "The Sum" from "Product No. 1," and call the answer "The Remainder."

Fifth, divide "The Remainder" by the number of pounds contained in the weight, and the quotient will give the distance from the fulcrum, in inches, the weight is required to be placed to allow the valve to rise at a given pressure per square inch.

Example.—Let $3\frac{1}{2}$ inches equal diameter of the valve.
Let .7854 equal a constant.
Let $3\frac{1}{2}$ inches equal distance of center of valve from the fulcrum.
Let 125 pounds equal given pressure per square inch
Let $7\frac{3}{8}$ pounds equal weight of valve and spindle.
Let $28\frac{3}{4}$ pounds equal weight of lever.
Let $18\frac{1}{8}$ inches equal distance of center of gravity of lever from the fulcrum.
Let $140\frac{5}{8}$ pounds equal weight of the weight.

Reducing the vulgar fractions to decimals we have them placed with their respective whole numbers in their proper positions, as shown in Fig. 175.

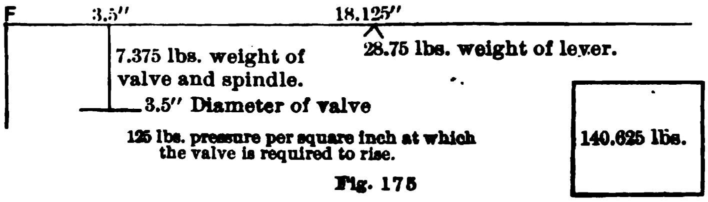


Fig. 175

Then, following the rule, we have:

$$\frac{(3.5\times3.5\times.7854\times3.5\times125)-\overline{(7.375\times3.5)+(28.75\times18.125)}}{140.625}=26.04+\text{ in.}$$

Distance the center of the weight is to be placed from the fulcrum.

Performing the operation in the ordinary way, we have:

3.5	Diameter of valve.
3.5	Diameter of valve.
1 75	
10	
12.25	Square of diameter of valve.
.7854	A constant.
4900	
6125	
9800	
8 575	
9.621150	Area of valve.
3.5	Distance of center of valve from fulcrum
4 8105750	
28 863450	
33.6740250	
125	Given pressure per square inch.
168 3701250	
673 480500	
3367 40250	
4209.2531250	"Product No. 1."

Next, multiplying the weight of the valve and spindle (7.375 pounds) by the distance of the center of the valve from the fulcrum (3.5 inches), we have:

7.375
3.5
3 6875
22 125
25.8125 "Product No. 2."

Next, multiplying the weight of the lever (28.75 pounds) by the distance of the center of gravity of the lever from the fulcrum (18.125 inches), we have:

28.75
18.125
14375
5750
2 875
230 00
287 5
521.09375 "Product No. 3."
Adding "Product No. 2" to "Product No. 3." 25.8125 "Product No. 2."
546.90625 "The Sum."

Next, subtracting "The Sum" from "Product No. 1," we have:

4209.253125 "Product No. 1."
546.90625 "The Sum."
3662.346875 "The Remainder."

Finally, dividing "The Remainder" by the number of pounds contained in the given weight, we have:

140.625000) 3662.346875 (26.04+ inches. Distance the center of weight is required to be placed from the fulcrum.
2812 50000
849 846875
843 750000
6 09687500
5 62500000

NUMBER OF POUNDS TO BE CONTAINED IN A WEIGHT TO BE PLACED A GIVEN DISTANCE FROM THE FULCRUM TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE IN POUNDS PER SQUARE INCH.

[Section 4418 of the Revised Statutes of the United States, and Section 5, of Rule 5, of the United States Board of Supervising Inspectors.]

RULE.—First, multiply the area of the valve by the distance of the center of the valve from the fulcrum; then multiply the product by the given pressure in pounds per square inch, and call the last product "Product No. 1."

Second, multiply the weight of the valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and call the product "Product No. 3."

Fourth, add Products Nos. 2 and 3 together, and call the answer "The Sum;" then subtract "The Sum" from "Product No. 1," and call the answer "The Remainder."

Fifth, divide "The Remainder" by the distance, in inches, the weight is to be placed on the lever from the fulcrum; the quotient will give the number of pounds to be contained in the weight.

Example.—Let $4\frac{1}{4}$ inches equal diameter of the valve.

Let .7854 equal a constant.

Let $4\frac{1}{2}$ inches equal distance of center of valve from the fulcrum.

Let 150 pounds per square inch equal given pressure.

Let $10\frac{1}{2}$ pounds equal weight of the valve and spindle.

Let $42\frac{1}{8}$ pounds equal weight of the lever.

Let $20\frac{3}{16}$ inches equal distance of center of gravity of lever from the fulcrum.

Let 40 inches equal distance of center of weight from the fulcrum.

Reducing the vulgar fractions to decimals we place them with their respective whole numbers, and the whole numbers not having any decimals, in their respective positions, as shown in the following diagram:

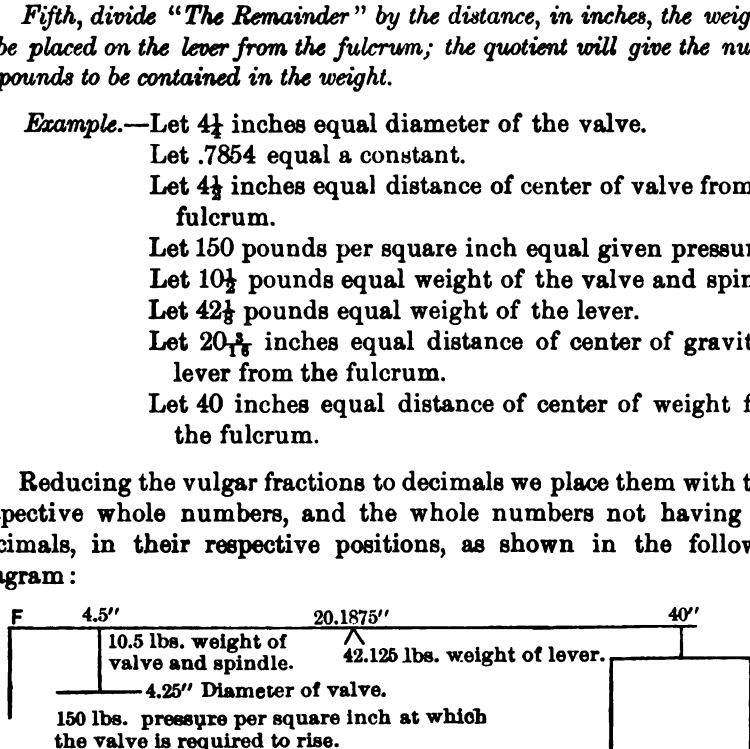


Fig. 176

Then we have:

$$\frac{(4.25\times4.25\times.7854\times4.5\times150)-\overline{(10.5\times4.5)+(42.125\times20.1875)}}{40}=216.95+ \text{ lbs. Weight required.}$$

Performing the operation in the ordinary way, we have:

4.25	Diameter of valve.
4.25	Diameter of valve.
2125	
850	
17 00	
18.0625	Square of diameter of valve.

Am't carried forward,

Am't brought forward,	18.0625	Square of diameter of valve.
	.7854	A constant.
	722500	
	903125	
	1 445000	
	12 64375	
	14.18628750	Area of valve.
	4.5	Distance of center of valve from fulcrum.
	7 093143750	
	56 74515000	
	63.838293750	
	150	Given pressure per square inch.
	3191 914687500	
	6383 8293750	
	9575.744062500	"Product No. 1."

Next, multiplying the weight of valve and spindle (10.5 pounds) by the distance of the center of the valve from the fulcrum (4.5 inches), we have:

10.5	
4.5	
5 25	
42 0	
47.25	"Product No. 2."

Next, multiplying the weight of the lever (42.125 pounds) by the distance of the center of gravity of the lever from the fulcrum (20.1875 inches), we have:

	42.125	
	20.1875	
	210625	
	294875	
	3 37000	
	4 2125	
	842 50	
	850.3984375	"Product No. 3."
Adding "Product No. 2" to "Product No. 3."	47.25	"Product No. 2."
	897.6484375	"The Sum."

Next, subtracting "The Sum" from "Product No. 1," we have:

9575.7440625	"Product No. 1."
897.6484375	"The Sum."
8678.0956250	"The Remainder."

Finally, dividing "The Remainder" by the distance, in inches, the center of the weight is to be placed on the lever from the fulcrum, we have:

```
40.000000) 8678.095625 (216.95+ lbs. Weight required.
           8000 0000
           ---------
            678 09562
            400 00000
            ---------
            278 095625
            240 000000
            ----------
             38 0956250
             36 0000000
             ----------
              2 09562500
              2 00000000
              ----------
```

SIZE OF A CUBIC CAST-IRON WEIGHT REQUIRED TO BE PLACED A GIVEN DISTANCE FROM THE FULCRUM TO ALLOW THE VALVE TO RISE AT A GIVEN PRESSURE PER SQUARE INCH.

[Section 4418 of the Revised Statutes of the United States; and Section 5, of Rule 5, of the United States Board of Supervising Inspectors.]

RULE.—First, multiply the area of the valve by the distance of the center of the valve from the fulcrum; then multiply the product by the given pressure in pounds per square inch, and call the last product "Product No. 1."

Second, multiply the weight of the valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and call the product "Product No. 3."

Fourth, add Products Nos. 2 and 3 together, and call the answer "The Sum;" then subtract "The Sum" from "Product No. 1," and call the answer "The Remainder."

Fifth, divide "The Remainder" by the distance, in inches, the weight is to be placed on the lever from the fulcrum; the quotient will give the number of pounds to be contained in the weight.

Sixth, divide the number of pounds to be contained in the weight by the constant .2607, and extract the cube root of the quotient; the answer will give the size, in inches, of a cubic weight required to be placed a given distance from the fulcrum to allow the valve to rise at a given pressure.

Example.—Let 4 inches equal diameter of the valve.
Let .7854 equal a constant.
Let 4 inches equal distance of center of valve from the fulcrum.
Let 150 pounds per square inch equal given pressure.

Let 10 pounds equal weight of the valve and spindle.
Let 40 pounds equal weight of the lever.
Let 20 inches equal distance of center of gravity of lever from the fulcrum.
Let 40 inches equal distance of center of weight from the fulcrum.
Let 2607 ten thousandths of a pound equal weight of a cubic inch of cast-iron.

Then we have:

$$\frac{(4\times4\times.7854\times4\times150)-\overline{(10\times4)+(40\times20)}}{40}=167.496 \text{ lbs. Weight required.}$$

Then $\sqrt[3]{167.496\div.2607}=8.6+$ inches. Dimensions of cubic weight required.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r l}
4 & \text{Diameter of valve.} \\
4 & \text{Diameter of valve.} \\
\hline
16 & \text{Square of diameter of valve.} \\
.7854 & \text{A constant.} \\
\hline
4\,7124 & \\
7\,854 & \\
\hline
12.5664 & \text{Area of valve.} \\
4 & \\
\hline
50.2656 & \\
150 & \text{Given pressure per square inch.} \\
\hline
2513\,2800 & \\
5026\,56 & \\
\hline
7539.8400 & \text{“Product No. 1.”}
\end{array}$$

Next, multiplying the weight of the valve and spindle (10 pounds) by their distance from the fulcrum (4 inches), we have:

$$\begin{array}{r l}
10 & \\
4 & \\
\hline
40 & \text{“Product No. 2.”}
\end{array}$$

Next, multiplying the weight of the lever (40 pounds) by the distance of the center of gravity of the lever from the fulcrum (20 inches), we have:

$$\begin{array}{r l}
40 & \\
20 & \\
\hline
800 & \text{“Product No. 3.”}
\end{array}$$

Next, adding Products Nos. 2 and 3 together, we have:

```
800   "Product No. 3."
 40   "Product No. 2."
———
840   "The Sum."
```

Next, subtracting the "The Sum" from "Product No. 1," we have:

```
7539.84   "Product No. 1."
 840      "The Sum."
———————
6699.84   "The Remainder."
```

Next, dividing "The Remainder" by the distance the center of the weight is to be placed from the fulcrum, we have:

```
40) 6699.84 (167.496   Number of pounds to be contained in the weight.
    40
    ——
    269
    240
    ———
     299
     280
     ———
      198
      160
      ———
       384
       360
       ———
        240
        240
        ———
```

Next, dividing the number of pounds contained in the weight by .2607, we have:

```
.2607) 167.4960 (642.485+   Number of cubic inches to be contained in the weight.
       156 42
       ——————
        11 076
        10 428
        ——————
          6480
          5214
          ————
          12660
          10428
          —————
           22320
           20856
           —————
            14640
            13035
            —————
```

Finally, extracting the cube root of the quotient, we have:

	642.485 (8.6+ inches. Cubic weight required.
8×8×8=	512
8×8×300=19200	130 485
8×6×30= . 1440	
6×6= 36	
20676	1240 56

Then a safety-valve adjusted according to the following diagram, will rise at a pressure of 150 pounds per square inch.

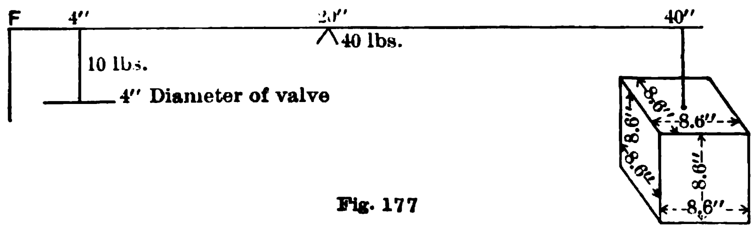


Fig. 177

PRESSURE REQUIRED TO RAISE A VALVE HAVING A SPHERICAL WEIGHT OF GIVEN DIAMETER ATTACHED TO LEVER.

As inspectors of marine boilers are often required to determine the pressure per square inch required to raise the valve attached to a steam boiler under pressure, the following diagrams will show how those valves are found in practice, and the rules will show how to determine the pressure per square inch required to raise them.

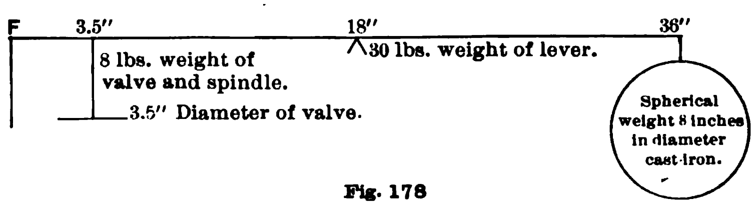


Fig. 178

RULE.—Determine the number of pounds contained in the spherical cast-iron weight by multiplying the cube of its diameter by the decimal .5236; *then multiply the product by* .2607, *and the last product will give the number of pounds contained in the weight.*

Thus:

8×8×8×.5236×.2607=69.889+ lbs. Weight of spherical weight.

Performing the operation in the ordinary way, we have:

8	Diameter of spherical weight
8	Diameter of spherical weight
64	Square of diameter of spherical weight.
8	Diameter of spherical weight.
512	Cube of diameter of spherical weight.
.5236	A constant.
1 0472	
5 236	
261 80	
268.0832	Number of cubic inches contained in the weight.
.2607	Weight of a cubic inch of cast-iron.
18765824	
16 084992	
53 61664	
69.88929024 pounds.	Weight of spherical weight.

We are now prepared to make the following diagram and proceed to determine the pressure per square inch required to raise the valve:

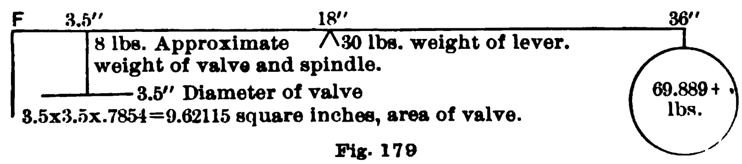


Fig. 179

RULE.—First, multiply the weight of the valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 1."

Second, multiply the weight of the lever by the distance of the center of gravity of the lever from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the weight by its distance from the fulcrum, and call the product "Product No. 3."

Fourth, add Products Nos. 1, 2 and 3 together, and call the answer "The Sum."

Fifth, multiply the area of the valve by the distance of its center from the fulcrum, and call the product "Product No. 4."

Sixth, divide "The Sum" by "Product No. 4," and the quotient will give the pressure per square inch required to raise the valve.

Example.—Let 8 lbs. equal weight of valve and spindle.

Let 3.5″ equal distance of center of valve from the fulcrum.

Let 30 lbs. equal weight of lever.

Let 18″ equal distance of center of gravity of lever from the fulcrum.

Let 69.889 lbs. equal weight of weight.

Let 36″ equal distance of centers of weight from the fulcrum.

Let 9.62115 square inches equal area of valve.

Then, we have:

$$\frac{(8\times3.5)+(30\times18)+(69.889\times36)}{9.62115\times3.5}=91.58+\text{ lbs.}$$ Pressure per square inch required to raise the valve.

Performing the operation in the ordinary way, we have:

3.5	Distance of center of valve from the fulcrum.
8	Weight of valve and spindle in pounds.
28.0	"Product No. 1."
30	Weight of lever.
18	Distance of center of gravity of lever from the fulcrum.
240	
30	
540	"Product No. 2."
69.889	Weight of weight.
36	Distance of center of weight from the fulcrum.
419 334	
2096 67	
2516.004	"Product No. 3."

Adding Products Nos. 1, 2 and 3 together, we have:

28	"Product No. 1."
540	"Product No. 2."
2516.004	"Product No. 3."
3084.004	"The Sum."

Next, multiplying the area of the valve by the distance of the center of the valve from the fulcrum, we have:

9.62115	Area of the valve.
3.5	Distance of center of valve from the fulcrum.
4 810575	
28 86345	
33.674025	"Product No. 4."

Finally, dividing "The Sum" by "Product No. 4," we have:

33.674025) 3084.004000 (91.58+ lbs. Pressure per square inch required to raise the valve.
3030 66225
53 341750
33 674025
19 6677250
16 8370125
2 83071250
2 69392200

It will be observed that the weight of the valve and spindle of a safety-valve attached to a steam boiler under pressure, can be obtained only by approximation, but the difference between the approximate weight of the valve and spindle and the actual weight of the same, can affect the result, at most, but a fractional part of a pound per square inch, and hence the rule in this case is sufficiently accurate to enable inspectors to determine whether the safety-valves have been tampered with, which is the real object of these rules.

Determine first the number of pounds contained in the weight.

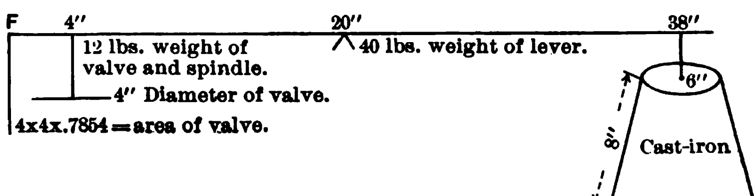


Fig. 180

PRESSURE REQUIRED TO RAISE VALVE WITH CAST-IRON CONIC WEIGHT OF GIVEN DIMENSIONS.

RULE.—First, subtract the diameter of the top from the diameter of the bottom, and divide the remainder by 2; then square the quotient, and call the product "Product No. 1."

Second, square the length of the side of the weight, and call the product "Product No. 2."

Third, subtract "Product No. 1" from "Product No. 2;" then extract the square root of the remainder, and the answer will give the length of the weight.

Fourth, multiply the diameter of the bottom by the diameter of the top, and add the product and the square of the top and the square of the bottom together, and call the answer "The Sum."

Fifth, multiply "The Sum" by .7854; then multiply the product by the length of the weight, and divide the last product by 3; the quotient will give the number of cubic inches contained in the weight.

Sixth, multiply the number of cubic inches contained in the weight by .2607, and the product will give the number of pounds contained in the weight.

Example.—Let 12″ equal diameter of bottom of weight.
Let 6″ equal diameter of top of weight.
Let 2 equal a constant.
Let 8″ equal length of the side of the weight.
Let .7854 equal a constant
Let 3 equal a constant.
Let 2607 ten thousandths of a pound equal weight of a cubic inch of cast-iron.

First, subtracting the diameter of the top from the diameter of the bottom, and dividing the remainder by 2, and squaring the quotient, we have:

```
                              12  Diameter of bottom of weight.
                               6  Diameter of top of weight.
                              ---
Dividing remainder by       2) 6  The remainder.
                              ---
                             { 3
Squaring the quotient.       { 3
                              ---
Square of the quotient.        9  "Product No. 1."
```

Second, squaring the length of the side of the weight, we have:

```
                               8  Length of side of weight.
                               8  Length of side of weight.
                              ---
Square of length of side of weight.  64  "Product No. 2."
```

Third, subtracting "Product No. 1" from "Product No. 2," and extracting the square root of the remainder, we have:

```
         64  "Product No. 2."
          9  "Product No. 1."
         ---
         55 (7.4162— inches.  Length through center of weight.
         49
        ----
    144) 600
         576
        -----
   1481) 2400
         1481
        ------
  14826) 91900
         88956
        -------
 148322) 294400
         296644
        -------
```

Fourth, multiplying the diameter of the bottom by the diameter of the top, and adding the product and the square of the diameter of the bottom together, we have:

12	Diameter of bottom of weight.
6	Diameter of top of weight.
72	Product of diameter of bottom multiplied by diameter of top.

Next we have:

6	Diameter of top of weight.
6	Diameter of top of weight.
36	Square of diameter of top of weight.

Next we have:

12	Diameter of bottom of weight.
12	Diameter of bottom of weight.
24	
12	
144	Square of diameter of bottom of weight.

Then we have the three products added together, thus:

72	Product of diameter of bottom multiplied by top of weight.
36	Square of diameter of top of weight.
144	Square of diameter of bottom of weight.
252	"The Sum."

Fifth, multiplying "The Sum" by .7854, and multiplying the product by the length of the weight, and dividing the last product by 3, we have:

```
        252
      .7854
     ------
       1008
      1 260
     20 16
    1764
    -------
   197.9208
     7.4162
   --------
    3958416
  1 1875248
  1 979208
 79 16832
1385 4456
-------------
3) 1467.82023696
-------------
  489.2734+   Number of cubic inches contained in the weight.
```

Sixth, multiplying the number of cubic inches contained in the weight by .2607, we have:

$$\begin{array}{r} 489.2734 \\ .2607 \\ \hline 34249138 \\ 29\,356404 \\ 97\,85468 \\ \hline 127.55357538 \end{array}$$

pounds. **Weight of conic-shaped weight.**

Having ascertained the number of pounds contained in the weight, we are enabled to complete the diagram preparatory to determining the pressure per square inch required to raise the valve.

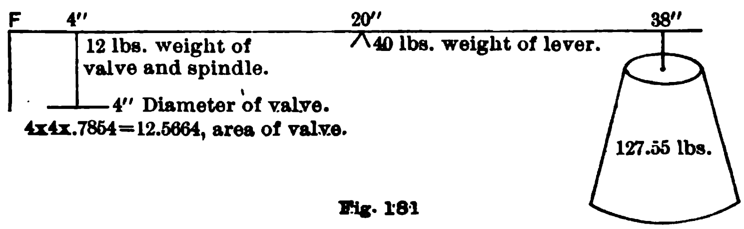


Fig. 181

We now proceed according to the rule laid down in the preceding case of the valve with spherical weight.

Example.—Let 12 lbs. equal weight of valve and spindle.
Let 4″ equal distance of center of valve from the fulcrum.
Let 40 lbs. equal weight of lever.
Let 20″ equal distance of center of gravity of lever from the fulcrum.
Let 127.55 lbs. equal weight of weight.
Let 12.5664 square inches equal area of valve.

Then we have:

$$\frac{(12\times4)+(40\times20)+(127.55\times38)}{12.5664\times4}=113+\text{ lbs.}$$

Pressure per square inch required to raise the valve.

Performing the operation in the ordinary way, we have:

$12\times 4=$	48	"Product No. 1."
$40\times20=$	800	"Product No. 2."
$127.55\times38=$	4846.9	"Product No. 3."
	5694.9	"The Sum."

Then, multiplying the area of the valve by the distance of its center from the fulcrum, and dividing "The Sum" by the product, we have:

12.5664×4= 50.2656) 5694.9000 (113+ lbs. Pressure per square inch required to raise the valve.
5026 56
668 340
502 656
165 6840
150 7968

PRESSURE REQUIRED TO RAISE A VALVE WITH TWO OR MORE CAST-IRON WEIGHTS OF GIVEN NUMBER OF POUNDS IN EACH.

Safety-valves with two or more weights, as shown by the following diagram, are frequently found in practice.

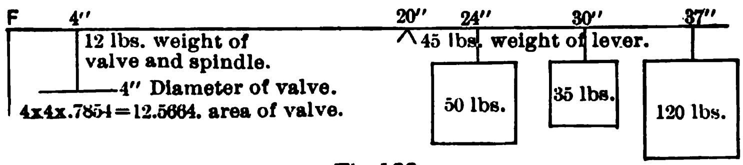


Fig. 182

RULE.—First, multiply the weight of valve and spindle by the distance of the center of the valve from the fulcrum, and call the product "Product No. 1."

Second, multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and call the product "Product No. 2."

Third, multiply the weight of the first weight by its distance from the fulcrum, and call the product "Product No. 3."

Fourth, multiply the weight of the second weight by its distance from the fulcrum, and call the product "Product No. 4."

Fifth, multiply the weight of the last weight by its distance from the fulcrum, and call the product "Product No. 5."

Sixth, add "Products Nos. 1, 2, 3, 4 and 5 together, and call the answer "The Sum."

Seventh, multiply the area of the valve by the distance of its center from the fulcrum, and call the product "Product No. 6."

Eighth, divide "The Sum" by "Product No. 6," and the quotient will give the pressure per square inch required to raise the valve.

Example.—Let 12 lbs. equal weight of valve and spindle.

Let 4″ equal distance of center of valve from the fulcrum.

Let 45 lbs. equal weight of lever.

Let 20″ equal distance of center of gravity of lever from the fulcrum.

Let 50 lbs. equal weight of first weight.

Let 24″ equal distance of first weight from the fulcrum.

Let 35 lbs. equal weight of second weight.

Let 30″ equal distance of second weight from the fulcrum.

Let 120 lbs. equal weight of third or last weight.

Let 37″ equal distance of third or last weight from the fulcrum

Let 12.5664 square inches equal area of the valve.

Then we have:

$$\frac{(12\times4)+(45\times20)+(50\times24)+(35\times30)+(120\times37)}{12.5664\times4}=151.95+\text{ lbs.}$$

Pressure per square inch required to raise the valve.

Performing the operation in the ordinary way, we have:

First,	$12\times 4 = 48$	"Product No. 1."
Second,	$45\times 20 = 900$	"Product No. 2"
Third,	$50\times 24 = 1200$	"Product No. 3."
Fourth,	$35\times 30 = 1050$	"Product No. 4."
Fifth,	120×37 4440	"Product No. 5."
Sixth,	7638	"The Sum."
Seventh,	$12.5664\times 4 = 50.2656$	"Product No. 6."

Finally, dividing "The Sum" by "Product No. 6," we have:

```
50.2656) 7638.0000 (151.95+ lbs.
         5026 56
         -------
         2611 440
         2513 280
         --------
           98 1600
           50 2656
           --------
           47 89440
           45 23904
           --------
            2 655360
            2 513280
           ---------
```

Pressure per square inch required to raise the valve.

THE UNITED STATES LEVER SAFETY-VALVE.

Fig. 183 Fig. 185

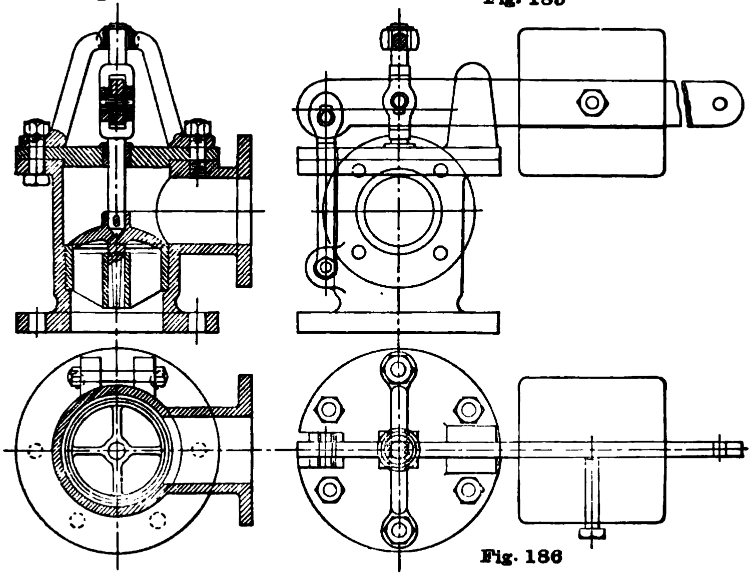


Fig. 186

Fig. 184

SPECIFICATIONS FOR UNITED STATES LEVER SAFETY-VALVE.

Section 24, of Rule 2, of the United States Board of Supervising Inspectors provides, that all lever safety-valves to be attached to marine boilers must be made according to the following specifications:

All lever safety-valves, to be attached to marine boilers, shall have an area of not less than one square inch to two square feet of grate surface of the boiler or boilers to which they are attached; and the seats of all such valves shall have an angle of inclination of 45 degrees to the center line of their axis.

All the points of bearing on the lever must be in the same plane.

The distance of the fulcrum from the bearing of the lever on the spindle directly over the center of the valve must in no case be less than the diameter of the valve opening.

The length of the lever should not exceed the distance of the fulcrum from the center of the valve multiplied by 10.

The width of the bearings of the fulcrum must not be less than three-fourths of one inch.

The length of the fulcrum links should not be less than 4 inches.

The lever or fulcrum points must be made of wrought iron or steel; and the knife-edge fulcrum points and bearings for the points must be made of steel and hardened.

The valve seat and bushings for the valve spindle must be made of composition (gun metal) when the valve is intended to be attached to a boiler using salt water; but when the valve is intended to be attached to a boiler using fresh water, and generating steam of a high pressure, the parts named, with the exception of the bushings for the spindle, may be made of cast-iron.

The valve must be guided by its spindle, both above and below the ground seat, and above the lever, through supports made of composition (gun metal), or bushed with it.

The spindle should fit loosely in the bearings or supports.

When the valve is intended to be attached to the boilers of steamers navigating rough waters, the fulcrum links may be connected directly with the spindle of the valve; provided always that the knife-edged fulcrum points are made of steel and hardened, and that the vertical movement of the valve is unobstructed by any lateral movement. In this case the short end of the lever should be attached directly to the valve casing.

SPRING-LOADED SAFETY-VALVES.

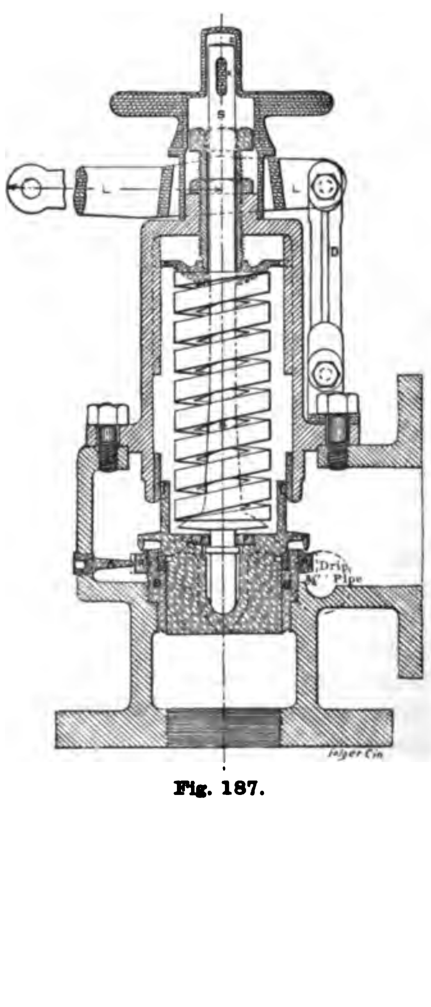
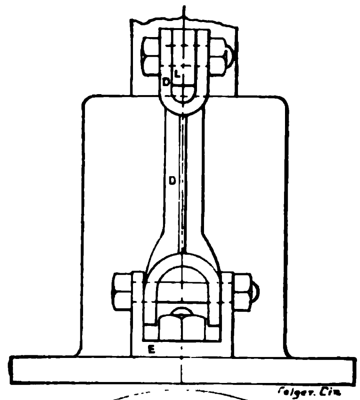


Fig. 187.

The engravings represent a spring-loaded valve, made in accordance with the requirements of the United States laws. Quite a number of this kind of valves are made by different manufacturers, all differing in some degree from each other, and yet made in compliance

Fig. 188.



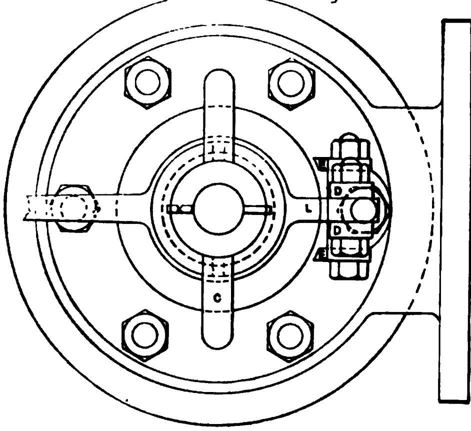


Fig. 189.

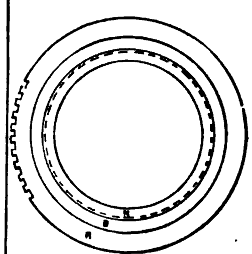


Fig. 190.

with the requirements of the law. It is provided with an overhanging lip, the effect of which is to increase the area presented to the operation of the steam after the valve is raised from its seat, and cause an increased lift in the movement of the valve from its seat. This movement is sudden, and if the valve was held in position by a lever and weight instead of a spring, the tendency would be to break the lever the first time the valve raised from its seat.

Fig. 187 shows the valve and parts in vertical section. Fig. 188 shows the arrangement of the link D to which the lever L (Figs. 187 and 189) is attached; which lever is employed to raise the valve from

its seat, as required by law. Fig. 190 is a plan showing the brass bush B, with nickel seat N, and adjustable ring R, all of which are also shown in cross section in Fig. 187. The plan of the adjustable ring shows the vertical notches around its circumference, and by which it is turned up or down on the bush B, as shown in Fig. 187, by the medium of any sharp-pointed instrument, inserted from the outside of the valve case through the hole now occupied by the pin A, which pin is merely used to hold the ring in place after it has been properly adjusted. The object of the adjustable ring is to increase or diminish the size of the steam-escape orifice around the circumference of the inside of the overhanging lip of the valve head, and which is accomplished by screwing the ring up or down on the bush B; the reason for this regulation is to adjust the closing point so as to prevent too great a reduction of pressure in the boiler before the valve seats. By screwing the adjustable ring upward the seating of the valve is prolonged, and the steam pressure can be made to fall considerably below the point at which the valve was forced from its seat. By screwing the adjustable ring downward the valve can be made to seat within a fraction of the pressure required to raise it. It is good practice, however, to adjust the ring so as to allow the valve to seat within two or three pounds of the pressure required to raise it.

The bearing of the lower end of the spindle S, it will be observed, is considerably below the seat of the valve; this is to prevent tipping of the valve and cause it to come fairly to its seat after being lifted or raised from it. The spindle near the lower end is provided with a collar, by which the valve may be forced from its seat should it become fast or stick to the seat; this is also a requirement of the law, because oftentimes safety-valves become fast and stick to their seats, and no lifting of the spindle will cause them to rise. It is, therefore, provided by law that all safety-valves shall be so constructed that they can be forced from their seats by means of levers, or other suitable arrangements. The United States marine lever safety-valve is provided with a pin which passes through the lower end of the spindle and through a hub on the valve containing the lower end of the spindle.

DIMENSIONS OF STEAM DRUM LEGS.

Section 4435, of the Revised Statutes of the United States provides, that where two or more boilers have water connections, "there shall also be provided a similar steam connection, having an area of opening into each boiler of at least one square inch for every two square feet of effective heating surface contained in any one of the boilers so connected, half the flue and all other fire surface being computed as effective."

DIAMETER OF STEAM DRUM LEGS OR OPENING INTO EACH BOILER.

RULE.—Divide the number of square feet of effective heating surface by 2; *then divide the quotient by* .7854, *and extract the square root of the last quotient.*

Example.—Let 44 inches equal diameter of the boilers.
Let 3.1416 equal a constant.
Let two-thirds of circumference of boilers equal portion exposed to heat.
Let 30 feet equal length of boilers.
Let 2 equal number of flues in each boiler.
Let 16 inches equal diameter of each flue.
Let 144 square inches equal number of square inches in a square foot.

First, determine the number of square feet of effective heating surface.

RULE.—First, multiply the constant 3.1416 *by the diameter of the boiler, in inches; then divide the product by* 3, *and multiply the quotient by* 2; *the last product will give two-thirds of the circumference of the boiler.*

Second, multiply the length of the boiler, in feet, by 12, *and the product will give the length of the boiler in inches.*

Third, multiply two-thirds of the circumference of the boiler, in inches, by the length, in inches, and the product will give the effective heating surface of the shell in square inches; the last product we will call "Product No. 1."

Fourth, as but half the inner surface of each flue is to be computed as effective, we multiply the constant 3.1416 *by the diameter of the flue in inches; the product will give the circumference of the flue.*

Fifth, multiply the length of the flue, in feet, by 12, *and the product will give the length of the flue in inches.*

Sixth, multiply the circumference of the flue, in inches, by the length, in inches, and the product will give the entire surface of one flue, or one-half of each of the two flues, in square inches.

Seventh, add the number of square inches in the shell of the boiler exposed to the heat to the number of square inches in the upper half of each flue in square inches, and the sum will give the entire number of square inches of effective heating surface.

Eighth, divide the entire effective heating surface in square inches by 144, *and the quotient will give the number of square feet of effective heating surface.*

Ninth, divide the number of square feet of heating surface by 2; *then divide the quotient by* .7854, *and extract the square root of the last quotient; the answer will give the diameter of the legs of the steam drum or opening into each boiler.*

Then we have:

First,

$3.1416 \times 44 = 138.2304$ inches. Circumference of boiler.

And,

$$\begin{array}{r} 3)\ \underline{138.2304} \\ 46.0768 \\ \underline{2} \\ 92.1536 \end{array}$$

92.1536 inches. Two-thirds of the circumference of boiler.

Second,

$30 \times 12 = 360$ inches. Length of boiler.

Third,

$$\begin{array}{r} 92.1536 \\ \underline{360} \\ 5529\ 2160 \\ \underline{27646\ 08} \\ 33175.2960 \end{array}$$

92.1536 Two-thirds of the circumference of the boiler in inches.
360 Length of boiler in inches.

33175.2960 square inches. Effective heating surface of shell of boiler.

Fourth,

$3.1416 \times 16 = 50.2656$ inches. Circumference of one flue, or two half flues.

Fifth,

$30 \times 12 = 360$ inches. Length of flue.

Sixth,

$$\begin{array}{r} 50.2656 \\ \underline{360} \\ 3015\ 9360 \\ \underline{15079\ 68} \\ 18095.6160 \end{array}$$

50.2656 Circumference of flue in inches.
360 Length of flue in inches.

18095.6160 square inches. Effective heating surface in both flues.

Seventh,

33175.2960 square inches. Effective heating surface in shell of boiler.
18095.6160 square inches. Effective heating surface in flues.

51270.9120 square inches. Total effective heating surface in boiler.

Eighth,

$$\begin{array}{l} 144)\ 51270.9120\ (356+ \\ \underline{432} \\ 807 \\ \underline{720} \\ 870 \\ \underline{864} \end{array}$$

(356+ square feet. Total heating surface of boiler.

Ninth, dividing the number of square feet of effective heating surface by 2, and then dividing the quotient by .7854, we have:

```
       2) 356
          ------------
   .7854) 178.0000 (226+   Last quotient.
          157 08
          -------
           20 920
           15 708
           -------
            5 2120
            4 7124
            -------
```

Finally, extracting the square root of the last quotient, we have:

```
       226 (15+ inches.   Required diameter of legs of
       1                  steam drum.
       ---
   25) 126
       125
       ---
```

BOILER FLUES.

[Rules based upon the requirements of the Rules and Regulations of the United States Board of Supervising Inspectors.]

To obviate extreme thickness of material in boiler flues, and avoid impairing the efficiency of the flues in the generation of steam, flues are made in sections in nearly all cases where high pressure of steam and long flues are required. In this manner the thickness of the material can be reduced to its minimum. To accomplish the same object various kinds of flues have been introduced; notably the corrugated and ribbed flues, which flues have become familiar to engineers and boiler makers.

Corrugated flues should have their ends plain not to exceed 6 inches, according to the United States Rules, and, for large flues, the corrugations should not be less than $1\frac{1}{2}$ inches deep, and the flues must conform practically to a true circle; and may be allowed a working steam pressure according to their thickness of material and diameter of flue without regard to length, as prescribed in Section 14. of Rule 2, of the Board of Supervising Inspectors.

SAFE-WORKING STEAM PRESSURE ALLOWABLE FOR CORRUGATED AND RIBBED FLUES.

RULE.—Divide the constant whole number 14,000 *by the mean diameter of the flue, in inches, and multiply the quotient by the thickness of material in decimals of an inch; the product will give the working steam pressure allowable.*

Example.—Let 14,000 equal a constant.

Let 40 inches equal mean diameter of the flue.

Let 5 tenths of an inch equal thickness of material in the flue.

Then we have:

$$\left(\frac{14000}{40}\right)\times .5 = 175 \text{ lbs.}$$ Pressure per square inch allowable.

Performing the operation in the ordinary way, we have:

```
40) 14000 (350
    120      .5  Thickness of material.
    ---     ----
    200  175.0 lbs. Pressure per square inch allowable.
    200
    ---
      0
```

THICKNESS OF MATERIAL REQUIRED IN CORRUGATED FLUES FOR ANY REQUIRED PRESSURE PER SQUARE INCH.

RULE.—Multiply the pressure required per square inch by the mean diameter of the flue, in inches, and divide the last product by the constant whole number 14,000; *the quotient will give the thickness of material required in decimals of an inch.*

Example.—Let 175 pounds per square inch equal the required pressure.

Let 40 inches equal mean diameter of the flue.

Let 14,000 equal a constant.

Then we have:

$$\frac{175\times 40}{14000} = .5$$ Decimals of an inch. Thickness of material required.

Performing the operation in the ordinary way, we have:

```
          175  Pressure per square inch
           40  Diameter of the flue.
        ------
14000) 7000.0 (0.5  Decimals of an inch. Thickness of material required.
       7000 0
       ------
```

Ribbed flues, in regard to pressure per square inch allowable, and thickness of material required, are governed by the same rules that govern corrugated flues.

PLAIN CYLINDRICAL FURNACE FLUES.

[Section 15, of Rule 2, of the United States Board of Supervising Inspectors.]

PRESSURE ALLOWABLE FOR PLAIN CYLINDRICAL FURNACE FLUES.

RULE.—First, multiply the constant whole number 89,600 *by the square of the thickness of the material in the flue, in decimals of an inch, and call the last product "Product No.* 1."

Second, multiply the diameter, in inches, by the length of the flue, in feet, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the steam pressure allowable per square inch.*

Example.—Let 89,600 equal a constant.
Let 5 tenths of an inch equal thickness of material.
Let 40 inches equal diameter of the flue.
Let 7 feet equal the length of the flue.

Then we have:

$$\frac{89600 \times .5 \times .5}{40 \times 7} = 80 \text{ lbs.}$$ Steam pressure per square inch allowable.

Performing the operation in the ordinary way, we have:

	89600	A constant.
.5×.5=	.25	Square of thickness of material.
	448000	
	179200	
	22400.00	"Product No. 1."

Next we have:

40	Diameter of flue in inches.
7	Length of flue in feet.
280	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
280) 22400 (80 lbs. Pressure per square inch allowable.
     2240
     ----
        0
```

FURNACE FLUES WITH ADAMSON AND ANGLE-IRON RINGS.

Where plain cylindrical furnace flues are made in sections, and flanged to a depth of not less than 2½ inches, and substantially riveted together with wrought-iron rings between such flanges, and such rings having a thickness of not less than one-half inch and a width of not less

than 2½ inches; or in lieu thereof, angle-iron rings are employed, and such rings having a thickness of not less than double the thickness of material in the flue and a depth of not less than 2½ inches, and substantially riveted in position with wrought-iron thimbles between the

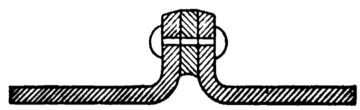


Fig. 191
ADAMSON RING.
FLANGED JOINT REQUIRED BY RULE 2, SECTION 15.

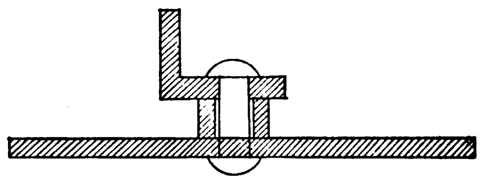


Fig. 192
ANGLE-IRON STRENGTHENING RING REQUIRED BY RULE 2, SECTION 15.

inner surface of such rings and the outer surface of the flue, at a distance from the flue not to exceed 2 inches, with rivets, having a diameter of not less than one and one-half times the thickness of material in the flue, and placed apart at a distance not to exceed 6 inches from center to center at the outer surface of the flue, the distance between the flanges or the distance between such angle-iron rings shall be taken as the length of the flue in determining the pressure allowable.

WORKING STEAM PRESSURE ALLOWABLE.

RULE.—First, multiply the constant whole number 89,600 *by the square of the thickness of material in the flue, in decimals of an inch, and call the product "Product No.* 1."

Second, multiply the diameter of the flue, in inches, by the distance between the flanges of the longest section, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the pressure per square inch allowable.*

Example.—Let 89,600 equal a constant.
Let 5 tenths of an inch equal thickness of material in the flue.
Let 40 inches equal diameter of the flue.
Let 5 feet equal distance between flanges or angle-iron rings.

Then we have:

$$\frac{89600\times .5\times .5}{40\times 5}=112 \text{ lbs.}$$ Pressure per square inch allowable.

Performing the operation, we have:

$$\begin{array}{r} .5 \\ .5 \\ \hline .25 \end{array}$$ Square of thickness of material in the flue.

Then, multiplying the constant whole number (89,600) by the square of the thickness of material in the flue (.25), we have:

$$\begin{array}{r} 89600 \\ .25 \\ \hline 4480\,00 \\ 17920\,0 \\ \hline 22400.00 \end{array}$$ "Product No. 1."

Next we have:

$$\begin{array}{r} 40 \\ 5 \\ \hline 200 \end{array}$$

40 Diameter of flue in inches.
5 Distance in feet between rings of longest section.
200 "Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

$$\begin{array}{l} 200)\,22400\,(112 \text{ lbs.} \\ 200 \\ \hline 240 \\ 200 \\ \hline 400 \\ 400 \\ \hline \end{array}$$ Pressure per square inch allowable.

THICKNESS OF MATERIAL REQUIRED FOR A PLAIN FURNACE FLUE OF A GIVEN DIAMETER, GIVEN LENGTH AND GIVEN STEAM PRESSURE PER SQUARE INCH.

RULE.—First, multiply the given diameter of the flue, in inches, by the length of the flue, in feet; then multiply the product by the given pressure per square inch, and call the last product "Product No. 1."

Second, divide "Product No. 1" by the constant whole number 89,600, *and extract the square root of the quotient; the answer will give the thickness of material in decimals of an inch.*

Example.—Let 40 inches equal diameter of the flue.
Let 4 feet equal length of the flue.
Let 140 pounds per square inch equal given pressure.
Let 89,600 equal a constant.

Then we have:

$$\sqrt{\frac{40\times4\times140}{89600}}=.5$$ Decimals of an inch. Thickness of material required.

Performing the operation in the ordinary way, we have:

40	Diameter of flue in inches.
4	Length of flue in feet.
160	
140	Pressure per square inch.
6400	
160	
22400	"Product No. 1."

Next, dividing "Product No. 1" by the constant 89,600, we have:

```
89600) 22400.0 (0.25   The quotient.
       17920 0
       -------
        4480 00
        4480 00
        -------
```

Finally, extracting the square root of the quotient, we have:

```
.25 (.5   Decimals of an inch. Thickness of material required.
.25
---
```

DIAMETER REQUIRED OF A FURNACE FLUE OF A GIVEN LENGTH, GIVEN THICKNESS OF MATERIAL AND GIVEN STEAM PRESSURE PER SQUARE INCH.

RULE.—First, multiply the constant whole number 89,600 *by the square of the thickness of material, in decimals of an inch, and call the last product "Product No.* 1."

Second, multiply the given pressure per square inch by the length of the flue in feet, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the required diameter of the flue in inches.*

Example.—Let 89,600 equal a constant.

Let 5 tenths of an inch equal thickness of material.
Let 140 pounds per square inch equal given pressure.
Let 4 feet equal length of the flue.

Then we have:

$$\frac{89600 \times .5 \times .5}{140 \times 4} = 40 \text{ inches. } \textbf{Required diameter of the flue.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rrl}
 & 89600 & \text{A constant.} \\
.5 \times .5 = & .25 & \text{Square of thickness of material.} \\
\hline
 & 4480\,00 & \\
 & 17920\,0 & \\
\hline
 & 22400.00 & \text{“Product No. 1.”}
\end{array}$$

Next we have:

$$\begin{array}{rl}
140 & \text{Given pressure per square inch.} \\
4 & \text{Length of flue in feet.} \\
\hline
560 & \text{“Product No. 2.”}
\end{array}$$

Finally, dividing "Product No. 1" by "Product No. 2," we have:

$$\begin{array}{l}
560)\ 22400\ (40 \text{ inches. } \textbf{Required diameter of the flue.} \\
2240 \\
\hline
0
\end{array}$$

LENGTH OF A FURNACE FLUE, OR LENGTH BETWEEN THE FLANGES OF SUCH A FLUE MADE IN SECTIONS, WHEN THE THICKNESS OF MATERIAL, STEAM PRESSURE PER SQUARE INCH, AND DIAMETER OF THE FLUE ARE GIVEN.

RULE.—First, multiply the constant whole number 89,600 *by the square of the thickness of material, in decimals of an inch, and call the product "Product No.* 1."

Second, multiply the given pressure per square inch by the diameter of the flue, in inches, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the required length of the flue, or longest section between the flanges, in feet.*

Example.—Let 89,600 equal a constant.
Let 5 tenths of an inch equal thickness of material.
Let 140 pounds per square inch equal given pressure.
Let 40 inches equal diameter of the flue.

Then we have:

$$\frac{89600 \times .5 \times .5}{140 \times 40} = 4 \text{ feet. } \textbf{Length of flue or longest section.}$$

Performing the operation in the ordinary way, we have:

	89600	A constant.
.5×.5=	.25	Square of thickness of material.
	4480 00	
	17920 0	
	22400.00	"Product No. 1"

Next we have:

140	Given pressure per square inch.
40	Diameter of the flue.
5600	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

5600) 22400 (4 feet. Length of flue or longest section.
22400

HALF-ROUND IRON STRENGTHENING RINGS FOR FLUES.

[Section 15, of Rule 2, of the United States Board of Supervising Inspectors.]

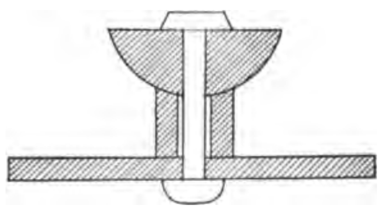


Fig. 193

HALF-ROUND IRON STRENGTHENING RING REQUIRED BY RULE 2, SECTION 15.

The strengthening rings may be made of half-round iron, as shown in Fig. 193, containing an area of cross section of not less than 9.6 times the thickness of material in the flue, and held in position around the flue with thimbles, at a distance from the surface of the flue not to exceed 2 inches, and substantially riveted with rivets, spaced not more than 8 inches from center to center at the surface of the flue, for rivets having a diameter of not less than seven-eighths of an inch; and not more than 6 inches from center to center for rivets having a diameter of not less than three-quarters of an inch; and not more than 4 inches from center to center for rivets having a diameter of not less than five-eighths of an inch; and no such rivets shall be allowed having a diameter of less than five-eighths of an inch.

REQUIRED DIAMETER OF THE HALF-ROUND IRON STRENGTHENING RING.

RULE.—Multiply the constant 9.6 *by the thickness of the material in the flue, in decimals of an inch, and multiply the product by* 2; *then divide the last product by* .7854, *and extract the square root of the quotient; the answer will give the required diameter of the half-round iron strengthening ring.*

Example.—Let 9.6 equal a constant.

Let 5 tenths of an inch equal thickness of material in the flue.

Let 2 equal a constant.

Let .7824 equal a constant.

Then we have: $\sqrt{\dfrac{9.6\times.5\times2}{.7854}}=3.5—$ inches. Required diameter of half-round iron strengthening ring.

Performing the operation in the ordinary way, we have:

```
            9.6    A constant.
             .5    Thickness of material.
           ----
           4.80
              2    A constant.
          ------
   .7854) 9.6000 (12.22+   The quotient.
          7 854
          ------
          1 7460
          1 5708
          ------
           17520
           15708
           ------
            18120
            15708
            -----
```

Next, extracting the square root of the quotient, we have:

```
         12.22 (3.5— inches.   Required diameter of half-round iron strengthening ring.
         9
        ----
    65) 322
        325
        ----
```

REQUIRED DIAMETER OF HALF-ROUND IRON STRENGTHENING RING—SIMPLE RULE.

RULE.—Multiply the constant 24.5 *by the thickness of material in the flue, in decimals of an inch, and extract the square root of the product.*

Example.—Let 24.5 equal a constant.

Let 5 tenths of an inch equal thickness of material in the flue.

Then we have: $\sqrt{24.5\times.5}=3.5$ inches. Required diameter of half-round iron ring.

Performing the operation in the ordinary way, we have:

```
   24.5   A constant.
     .5   Thickness of material.
  -----
  12.25   Product.
```

Next, extracting the square root of the product, we have:

$$\begin{array}{r l}
 & \dot{1}2.\dot{2}5\ (3.5 \text{ inches. Required diameter of half-round iron ring.}\\
 & 9\\
 \hline
 65) & 325\\
 & 325\\
 \hline
\end{array}$$

WORKING STEAM PRESSURE ALLOWABLE FOR FLUES HAVING HALF-ROUND IRON STRENGTHENING RINGS.

RULE.—First, multiply the constant whole number 89,600 by the square of the thickness of material in the flue, in decimals of an inch; then multiply the product by .75, and call the last product "Product No. 1."

Second, multiply the diameter of the flue, in inches, by the distance from center to center of strengthening rings, in feet, and call the product "Product No. 2."

Third, divide "Product No. 1" by "Product No. 2," and the quotient will give the working steam pressure per square inch allowable.

Example.—Let 89,600 equal a constant.

Let 5 tenths of an inch equal thickness of material in the flue.

Let .75 equal a co-efficient.

Let 40 inches equal diameter of the flue.

Let 3 feet equal distance from center to center of strengthening rings.

Then we have:

$$\frac{89600 \times .5 \times .5 \times .75}{40 \times 3} = 140 \text{ lbs.}$$ Working steam pressure per square inch allowable.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r r l}
 & 89600 & \text{A constant.}\\
.5 \times .5 = & .25 & \text{Square of thickness of material.}\\
\hline
 & 4480\,00 & \\
 & 17920\,0 & \\
\hline
 & 22400.00 & \\
 & .75 & \\
\hline
 & 1120\,0000 & \\
 & 15680\,000 & \\
\hline
 & 16800.0000 & \text{"Product No. 1."}
\end{array}$$

Next we have:

$$\begin{array}{r l}
40 & \text{Diameter of flue in inches.}\\
3 & \text{Length of flue in feet.}\\
\hline
120 & \text{"Product No. 2."}
\end{array}$$

Finally, dividing "Product No. 1" by "Product No. 2," we have:

```
120) 16800 (140 lbs.   Working steam pressure per
     120                 square inch allowable.
     ---
      480
      480
      ---
        0
```

THICKNESS OF MATERIAL REQUIRED FOR FLUES REINFORCED WITH HALF-ROUND IRON STRENGTHENING RINGS.

RULE.—First, multiply the given pressure, in pounds per square inch, by the diameter of the flue, in inches, and multiply the product by the distance between the strengthening rings of the flue, in feet, and call the last product "Product No. 1."

Second, multiply the constant 89,600 *by the co-efficient* .75, *and call the product "Product No. 2."*

Third, divide "Product No. 1" by "Product No. 2," and extract the square root of the quotient; the answer will give the required thickness of material in decimals of an inch.

Example.—Let 140 pounds equal given pressure per square inch.
Let 40 inches equal diameter of the flue.
Let 3 feet equal distance from center to center of strengthening rings.
Let 89,600 equal a constant.
Let .75 equal a co-efficient.

Then we have:

$$\sqrt{\frac{140\times40\times3}{89600\times.75}}=.5$$ Decimals of an inch. Required thickness of material for flue.

Performing the operation in the ordinary way, we have:

```
 140 lbs.      Given pressure per square inch.
  40 inches.   Diameter of flue.
 ----
5600
   3 feet.     Distance between strengthening rings.
 -----
16800          "Product No. 1."
```

Next we have:

```
  89600        A constant.
    .75        A co-efficient
 --------
 4480 00
62720 0
 --------
67200.00       "Product No. 2.
```

Next, dividing "Product No. 1" by "Product No. 2," we have:

```
67200) 16800.0 (0.25   The quotient.
       13440 0
       -------
        3360 00
        3360 00
        -------
```

Finally, extracting the square root of the quotient, we have:

```
.25 (.5   Decimals of an inch. Thickness of material required.
.25
---
```

DISTANCE FROM CENTER TO CENTER OF HALF-ROUND IRON STRENGTHENING RINGS.

RULE.—First, multiply the constant 89,600 *by the co-efficient* .75, *and then multiply the product by the square of the thickness of material in the flue, in decimals of an inch, and call the last product "Product No.* 1."

Second, multiply the given pressure, in pounds per square inch, by the diameter of the flue, in inches, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the distance from center to center of half-round iron strengthening rings.*

Example.—Let 89,600 equal a constant.
Let .75 equal a co-efficient.
Let 5 tenths of an inch equal thickness of material in the flue.
Let 140 pounds equal given pressure per square inch.
Let 40 inches equal diameter of the flue.

Then we have:

$$\frac{89600\times.75\times.5\times.5}{140\times40}=3\text{ feet.}$$ Distance from center to center of strengthening rings.

Performing the operation in the ordinary way, we have·

```
            89600        A constant.
              .75        A co-efficient.
          ---------
          4480 00
         62720 0
         ----------
         67200.00
5×.5=          .25       Square of the thickness of materi
         ----------
         3360 0000
        13440 000
        -----------
        16800.0000       "Product No. 1."
```

Next, multiplying the given pressure per square inch by the diameter of the flue, we have:

140 lbs.	Given pressure.
40 inches.	Diameter.
5600	"Product No. 2."

Finally, dividing "Product No. 1" by "Product No. 2," we have:

5600) 16800 (3 feet. Distance from center to center of strengthening rings.
16800

REQUIRED DIAMETER FOR A FLUE SUPPORTED WITH HALF-ROUND IRON STRENGTHENING RINGS.

RULE.—First, multiply the constant 89,600 *by* .75, *and then multiply the product by the square of the thickness of material in the flue, and call the last product "Product No.* 1."

Second, multiply the given pressure per square inch by the distance from center to center of reinforcement rings, in feet, and call the product "Product No. 2."

Third, divide "Product No. 1" *by "Product No.* 2," *and the quotient will give the required diameter of the flue in inches.*

Example.—Let 89,600 equal a constant.
Let .75 equal a co-efficient.
Let 5 tenths equal thickness of material in the flue.
Let 140 pounds equal given pressure per square inch.
Let 3 feet equal distance from center to center of reinforcement rings.

Then we have:

$$\frac{89600 \times .75 \times .5 \times .5}{140 \times 3} = 40 \text{ inches.}$$ Required diameter of flue.

Performing the operation in the ordinary way, we have:

	89600	A constant.
	.75	A co-efficient.
	448000	
	627200	
	6720000	
$.5 \times .5 =$	.25	Square of thickness of material.
	3360 0000	
	13440 000	
	16800.0000	"Product No. 1."

Next, multiplying the given pressure per square inch (140) by the distance from center to center of reinforcement rings in feet (3), we have:

$$\begin{array}{r} 140 \\ 3 \\ \hline 420 \end{array} \text{ "Product No. 2."}$$

Finally, dividing "Product No. 1" by "Product No. 2," we have:

$$\begin{array}{r} 420)\ 16800\ (40 \text{ inches. Required diameter of flue.} \\ 1680 \\ \hline 0 \end{array}$$

In all cases where the half-round iron ring is employed to strengthen flues, the area of cross section and diameter of such rings must be in accordance with the thickness of material in the flues, and such area of cross section, or diameter, must be determined by the first rule given under the head of "half-round iron strengthening rings for flues."

The reason for basing the area or diameter of such rings upon the thickness of material in the flues is that the steam pressure is based upon the thickness of material in the flue.

RIVETED AND LAP-WELDED FLUES MADE IN SECTIONS.

Section 8, of Rule 2, of the United States Board of Supervising Inspectors, provides that the following tables shall include all riveted and lap-welded flues exceeding 6 inches in diameter, and not exceeding 40 inches in diameter, not otherwise provided for by law. And all such flues shall be made in sections, according to their respective diameters, not to exceed the lengths prescribed in the table; and such sections shall be properly fitted one into the other, and substantially riveted, and the thickness of material required for any such flue of any given diameter shall, in no case, be less than the least thickness prescribed in the table for any such given diameter; and all such flues may be allowed the prescribed working steam pressure, if, in the opinion of the inspectors, it is deemed safe to make such allowance; and inspectors are therefore required, from actual measurement of each flue, to make such reduction from the prescribed working steam pressure for any material deviation in the uniformity of the thickness of material, or for any material deviation in the form of the flue from that of a true circle, as in their judgment, the safety of navigation may require.

A small amount of material has been added to the thickness of material for flues over 6, and not over 10, inches in diameter, in order to prevent flattening too easily when using the hammer to remove the scale. All other flues in the table are made according to rule:

TABLE OF DIMENSIONS REQUIRED AND PRESSURES ALLOWABLE.

SAFE-WORKING PRESSURE FOR CYLINDRICAL LAP-WELDED AND RIVETED BOILER FLUES MADE IN SECTIONS.

Greatest Length of Sections Allowable	5 FEET.				3 FEET.												
Least Thickness of Material Allowable	.18"	.20"	.21"	.21"	.22"	.22"	.23"	.24"	.25"	.26"	.27"	.28"	.29"	.30"	.31"	.32"	.33"
Diameters of Flues	Over 6" and not over 7"	Over 7" and not over 8"	Over 8" and not over 9"	Over 9" and not over 10"	Over 10" and not over 11"	Over 11" and not over 12"	Over 12" and not over 13"	Over 13" and not over 14"	Over 14" and not over 15"	Over 15" and not over 16"	Over 16" and not over 17"	Over 17" and not over 18"	Over 18" and not over 19"	Over 19" and not over 20"	Over 20" and not over 21"	Over 21" and not over 22"	Over 22" and not over 23"
Thickness of Material Required.	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure
.18 inch........	189	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.19 "	194	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.20 "	199	184	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.21 "	204	189	179	174	...	...	...	...	...	...	...	...	...	...	...	...	...
.22 "	...	194	184	179	172	158	...	...	...	...	...	...	...	...	...	...	...
.23 "	...	199	189	184	179	165	152	...	...	...	...	...	...	...	...	...	...
.24 "	...	204	194	189	187	172	158	147	...	...	...	...	...	...	...	...	...
.25 "	...	...	199	194	195	179	165	153	143	...	...	...	...	...	...	...	...
.26 "	...	...	204	199	203	186	172	159	148	139	...	...	...	...	...	...	...
.27 "	...	...	...	204	211	193	178	165	154	145	136	...	...	...	...	...	...
.28 "	...	...	...	...	...	200	185	172	160	150	141	134	...	...	...	...	...
.29 "	...	...	...	...	...	207	191	178	166	155	146	138	131	...	...	...	...
.30 "	...	...	...	...	...	...	198	184	172	161	152	143	135	129	...	...	...
.31 "	...	...	...	...	...	...	205	190	178	166	156	148	140	133	126	...	...
.32 "	...	...	...	...	...	...	...	196	183	171	161	153	145	137	131	125	...
.33 "	...	...	...	...	...	...	...	202	189	177	167	157	149	141	135	129	122
.34 "	...	...	...	...	...	...	...	...	194	182	172	162	154	146	139	132	126
.35 "	...	...	...	...	...	...	...	...	200	187	177	167	159	150	143	137	130
.36 "	...	...	...	...	...	...	...	...	...	193	182	172	163	154	147	141	134
.37 "	...	...	...	...	...	...	...	...	...	...	187	177	168	159	151	145	138
.38 "	...	...	...	...	...	...	...	...	...	...	...	182	172	163	155	149	142
.39 "	...	...	...	...	...	...	...	...	...	...	...	...	176	167	159	152	146
.40 "	...	...	...	...	...	...	...	...	...	...	...	...	...	172	163	156	150
.41 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	167	160	153
.42 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	164	157
.43 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	160

Table of Dimensions Required and Pressures Allowable.—Continued.

SAFE-WORKING PRESSURE FOR CYLINDRICAL LAP-WELDED AND RIVETED BOILER FLUES MADE IN SECTIONS.

Greatest Length of Sections Allowable	30 INCHES.																
Least Thickness of Material Allowable	.34″	.35″	.36″	.37″	.38″	.39″	.40″	.41″	.42″	.43″	.44″	.45″	.46″	.47″	.48″	.49″	.50″
Diameters of Flues	Over 23″ and not over 24″	Over 24″ and not over 25″	Over 25″ and not over 26″	Over 26″ and not over 27″	Over 27″ and not over 28″	Over 28″ and not over 29″	Over 29″ and not over 30″	Over 30″ and not over 31″	Over 31″ and not over 32″	Over 32″ and not over 33″	Over 33″ and not over 34″	Over 34″ and not over 35″	Over 35″ and not over 36″	Over 36″ and not over 37″	Over 37″ and not over 38″	Over 38″ and not over 39″	Over 39″ and not over 40″
Thickness of Material Required.	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure	Pounds Pressure
.34 inch........	121	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.35 "	125	120	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.36 "	129	123	119	...	...	...	...	...	...	...	...	...	...	...	...	...	...
.37 "	132	127	122	117	...	...	...	...	...	...	...	...	...	...	...	...	...
.38 "	136	130	125	121	116	...	...	...	...	...	...	...	...	...	...	...	...
.39 "	139	133	129	124	119	115	...	...	...	...	...	...	...	...	...	...	...
.40 "	143	137	132	127	123	118	115	...	...	...	...	...	...	...	...	...	...
.41 "	147	141	135	130	126	121	118	114	...	...	...	...	...	...	...	...	...
.42 "	151	144	139	133	129	124	120	117	112	...	...	...	...	...	...	...	...
.43 "	154	147	142	137	132	127	123	119	115	112	...	...	...	...	...	...	...
.44 "	157	151	145	140	135	130	126	122	118	115	110	...	...	...	...	...	...
.45 "	...	155	148	143	138	133	129	124	121	117	113	110	...	...	...	...	...
.46 "	...	...	151	146	141	136	132	127	124	120	116	112	109	...	...	...	...
.47 "	...	...	...	149	144	139	135	130	126	122	118	115	112	109	...	...	...
.48 "	...	...	...	...	147	142	138	133	129	125	120	117	114	111	108	...	...
.49 "	...	...	...	...	...	145	140	135	131	127	123	120	116	113	111	108	...
.50 "	...	...	...	...	...	...	143	138	134	130	126	122	119	116	113	110	107
.51 "	...	...	...	...	...	...	...	141	137	133	129	125	121	118	115	112	109
.52 "	...	...	...	...	...	...	...	...	140	135	131	127	123	120	117	114	111
.53 "	...	...	...	...	...	...	...	...	...	138	134	129	126	123	119	116	113
.54 "	...	...	...	...	...	...	...	...	...	...	137	132	128	125	122	118	116
.55 "	...	...	...	...	...	...	...	...	...	...	...	135	131	127	124	121	118
.56 "	...	...	...	...	...	...	...	...	...	...	...	...	133	130	126	123	120
.57 "	...	...	...	...	...	...	...	...	...	..	...	...	...	132	128	125	122
.58 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	131	127	124
.59 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	130	126
.60 "	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	129

SAFE-WORKING PRESSURE FOR ANY DIAMETER OF FLUE OVER 10 INCHES.

RULE.—Multiply the thickness of material, in hundredths of an inch, by the constant whole number 43, and divide the product by one-half of the diameter of the flue, in inches; the quotient will give the pressure per square inch allowable. Thickness of material to be used as whole numbers.

Example.—Let 30 equal the number of hundredths of an inch in thickness of material.
Let 43 equal a constant.
Let 16 inches equal diameter of flue.

Then we have: $\frac{30\times43}{16\div2}=161+$ lbs. Pressure per square inch allowable.

Performing the operation in the ordinary way, we have:

$$\begin{array}{r l}
30 & \text{Thickness of material.} \\
43 & \text{A constant.} \\
\hline
90 & \\
120 & \\
\hline
16\div2= \quad 8)\,1290 & \\
\hline
161+ & \text{lbs. Pressure per square inch allowable.}
\end{array}$$

INCREASE IN LENGTH OF SECTIONS ALLOWABLE BY REDUCTION IN PRESSURE

Riveted and lap-welded flues of any thickness of material, diameter and length of sections prescribed in the table, may be made in sections of any desired length, exceeding the maximum length allowed by the table, by reducing the prescribed pressure in proportion to the increased length of section according to the following rule:

RULE.—Multiply the pressure in the table allowed for any prescribed thickness of material and diameter of flue by the greatest length, in feet, of sections allowable for such flue, and divide the product by the desired length of sections, in feet, from center line to center line of rivets in the circular seams of such sections; the quotient will give the working steam pressure allowable.

Example.—Taking a flue in the table, 24 inches in diameter, required to be made in sections not exceeding 2.5 feet in length, and having a thickness of material of 44 one hundredths of an inch, and allowed a pressure of 157 pounds per square inch, and it is desired to make this flue in sections 5 feet in length.

Then we have: $\frac{157\times2.5}{5}=78.5$ lbs. Pressure allowable.

Performing the operation in the ordinary way, we have:

$$
\begin{array}{rrl}
 & 157 & \text{Given pressure.} \\
 & 2.5 & \text{Given length of sections.} \\
\hline
 & 78\,5 & \\
 & 314 & \\
\hline
\text{Dividing by desired length of sections.} \quad 5) & 392.5 & \\
\hline
 & 78.5 & \text{lbs. Pressure allowable.}
\end{array}
$$

THICKNESS OF MATERIAL REQUIRED FOR FLUES AND TUBES NOT OTHERWISE PROVIDED FOR.

[Section 9, of Rule 2, of the United States Board of Supervising Inspectors.]

Tubes and flues not exceeding 6 inches in diameter, and made of any prescribed length, and lap-welded flues required to carry a working steam pressure not to exceed 60 pounds per square inch, and having a diameter not exceeding 16 inches, and a length not exceeding 18 feet; and lap-welded flues required to carry a steam pressure exceeding 60 pounds per square inch, and not exceeding 120 pounds per square inch, and having a diameter not exceeding 16 inches, and a length not exceeding 18 feet, and made in sections not exceeding 5 feet in length, and fitted properly one into the other and substantially riveted; and all such tubes and flues shall have a thickness of material according to their respective diameters, as prescribed in the following table:

Outside Diameter, Inches.	Thickness of Material, Inch.	Outside Diameter, Inches.	Thickness of Material, Inch.	Outside Diameter, Inches.	Thickness of Material, Inch.
1	.072	3¼	.120	9	.180
1¼	.072	3½	.120	10	.203
1½	.083	3¾	.120	11	.220
1¾	.095	4	.134	12	.229
2	.095	4½	.134	13	.238
2¼	.095	5	.148	14	.248
2½	.109	6	.165	15	.259
2¾	.109	7	.165	16	.270
3	.109	8	.165	..	....

LAP-WELDED FLUES.

LAP-WELDED FLUES NOT EXCEEDING 6 INCHES IN DIAMETER.

[Section 10, of Rule 2, of the United States Board of Supervising Inspectors.]

Lap-welded flues not exceeding 6 inches in diameter may be made of any required length without being made in sections. And all such lap-welded flues and riveted flues not exceeding 6 inches in diameter, may be allowed a working steam pressure not to exceed 225 pounds per square inch, if deemed safe by the inspectors.

LAP-WELDED FLUES EXCEEDING 6 INCHES IN DIAMETER AND NOT EXCEEDING 16 INCHES IN DIAMETER AND NOT EXCEEDING 18 FEET IN LENGTH.

[Section 11, of Rule 2, of the United States Board of Supervising Inspectors.]

Lap-welded flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter and not exceeding 18 feet in length, and required to carry a steam pressure not exceeding 60 pounds per square inch, shall not be required to be made in sections.

[Section 12, of Rule 2, of the United States Board of Supervising Inspectors.]

Lap-welded and riveted flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter and not exceeding 18 feet in length, and required to carry a steam pressure exceeding 60 pounds per square inch and not exceeding 120 pounds per square inch, may be allowed, if made in sections 5 feet in length, and properly fitted, one into the other and substantially riveted.

LAP-WELDED AND RIVETED FLUES EXCEEDING 6 INCHES IN DIAMETER AND NOT EXCEEDING 40 INCHES IN DIAMETER.

[Section 13, of Rule 2, of the United States Board of Supervising Inspectors.]

Riveted and lap-welded flues exceeding 6 inches in diameter and not exceeding 40 inches in diameter, required to carry a working steam pressure per square inch exceeding the maximum steam pressure prescribed for any such flue in the table of Section 8, of this rule, shall be constructed under the provisions of Section 15, of this rule (Rule 2), and limited to the working steam pressure therein provided for furnace flues; but in no case shall the material in any such riveted or lap-welded flue be of less thickness for any given diameter than the least thickness prescribed in the aforementioned table for flues of such diameters.

COPPER STEAM PIPES.

THICKNESS OF MATERIAL REQUIRED.

[Section 38, of Rule 2, of the United States Board of Supervising Inspectors.]

All copper steam pipes shall be flanged to a depth of not less than four times the thickness of the material in the pipes, and all such flanging shall be made to a radius not to exceed the thickness of material in such pipes. And all such pipes shall have a thickness of material according to the working steam pressure allowed on the boilers, and such thickness of material shall be determined by the following rule:

RULE.—Multiply the working steam pressure allowed the boiler, in pounds per square inch, by the diameter of the pipe, in inches; then divide the product by the constant whole number 8000, and add .0625 to the quotient; the sum will give the thickness of material required in decimals of an inch.

Example.—Let 175 pounds equal working steam pressure per square inch allowed the boiler.
Let 5 inches equal diameter of the pipe.
Let 8000 equal a constant.
Let .0625 equal a constant.

Then we have:

$$\frac{175 \times 5}{8000} + .0625 = .1718+ \quad \text{Decimals of an inch. Thickness of material required.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl}
175 & \text{Steam pressure.} \\
5 & \text{Diameter of pipe.} \\
\hline
8000)\ 875.0\ (0.1093+ & \text{The quotient.} \\
800\ 0 & \\
\hline
75\ 000 & \\
72\ 000 & \\
\hline
3\ 0000 & \\
2\ 4000 & \\
\hline
\end{array}$$

Next, adding the constant .0625 to the quotient, we have:

$$\begin{array}{rl}
.1093 & \\
.0625 & \\
\hline
.1718 & \text{Decimals of an inch. Thickness of material required.}
\end{array}$$

UNITED STATES MARINE EXAMINATION QUESTIONS AND ANSWERS.

FOR ENGINEERS AND INSPECTORS.

Write application for appointment to the position of local inspector of boilers of steam vessels at.....................

In which give your name in full, age, native or naturalized citizen, experience, if any, in the construction of steam machinery, and in the care and management of such; and such other experience and knowledge required by Section 4415 of the Revised Statutes of the United States, and the qualifications prescribed by it and Section 4416 Revised Statutes.

Question. Reduce the following vulgar fractions to decimal fractions: $\frac{3}{8}$ $\frac{5}{8}$ $\frac{7}{8}$ $\frac{1}{16}$ $\frac{15}{16}$ $\frac{7}{32}$ $\frac{29}{32}$ $\frac{19}{64}$ $\frac{1}{64}$

Answer. .375 .625 .875 .0625 .9375 .21875 .90625 .296875 .015625

Q. Extract the square root of 3429904.

A. 1852.

Q. Extract the square root of 3461460.25.

A. 1860.5.

Q. Extract the cube root of 6486889625.

A. 1865.

Q. Extract the cube root of 6691435355.584.

A. 1884.4.

Q. Define the term "tensile strength" as employed in Section 3, of Rule 1, of the Board of Supervising Inspectors. In other words, what is meant by the tensile strength of boiler plate?

A. Tensile strength is the adhesive quality of the plate tending to resist a strain to separate it; and it means the total strain, in pounds, required to separate a square inch of cross section of the material.

Q. How is the tensile strength of boiler plate determined?

A. Tensile strength in boiler plate is determined by breaking a sample in a machine made for that purpose, and dividing the strain at which the sample parted by the area of sample at point of fracture before breaking.

Example.—Let .26 equal thickness of sample in hundredths of an inch.
Let .96 equal width of sample in hundredths of an inch.
Let 14,976 equal breaking strain in pounds.

Then we have:

```
     .26
     .96
    ----
     156
    234
   -----
   .2496) 14976.0000 (60000 lbs.
          14976
          ---------
              0000
```

Q. How is the ductility of boiler plate determined?

A. The ductility is obtained by subtracting the area of sample at point of fracture, after breaking, from the area of sample, before breaking, and dividing the remainder by the area of sample before breaking.

Example.—Let $.26 \times .96$ equal area of sample before breaking.
Let $.18 \times .80$ equal area of sample after breaking.

Then we have:

$.26 \times .96 = .2496$
$.18 \times .80 = .1440$

.2496).1056.0 (.42+ **Percentage of ductility.**
998 4
57 60
49 92

Q. Determine the bursting pressure per square inch of a boiler having a diameter of 42 inches and a thickness of material of .25 of an inch, made of wrought iron plates having a tensile strength of 55,000 pounds per square inch, with longitudinal seams double riveted and holes drilled? [Section 4415 Revised Statutes.]

A. Multiply the thickness of the weakest plate in the shell of the boiler by its tensile strength per square inch, and divide the product by one-half of the diameter of the boiler, in inches, and multiply the quotient by .70 for double-riveted longitudinal seams; by .65 for staggered-riveted longitudinal seams; and .56 for single-riveted longitudinal seams.

Example.—Let .25 equal thickness of weakest plate in hundredths of an inch.
Let 55,000 equal tensile strength per square inch of weakest plate.
Let 42 equal diameter of the boiler in inches.

Then we have:

.25
55000
125000
125
21) 1375000 (654.76+ lbs. **Would be the bursting pressure if there were no seams in the boiler.**
126
115
105
100
84
160
147
130
126

As the seams in this boiler are double-riveted we multiply the above quotient by .70, and we have the following:

$$\begin{array}{r} 654.76 \\ .70 \\ \hline 458.3320 \end{array}$$ lbs. Bursting pressure.

Q. Determine the bursting pressure per square inch of a boiler having a diameter of 48 inches, and a thickness of material of .26 of an inch, made of homogeneous steel plates having a tensile strength of 60,000 pounds per square inch, with longitudinal seams single riveted and holes drilled? [Section 4415 Revised Statutes.]

A. The rule for this is the same as that for the preceding question, only .56 is used as a constant instead of .70

Example.—Let .26 equal thickness of weakest plate in hundredths of an inch.

Let 60,000 equal tensile strength of weakest plate per square inch.

Let 48 inches equal diameter of boiler in inches.

Then we have:

$$\begin{array}{r} .26 \\ 60000 \\ \hline 24)\ 15600.00\ (650 \text{ lbs.} \\ 144 \\ \hline 120 \\ 120 \\ \hline \end{array}$$

Would be the bursting pressure if there were no seams in the shell of the boiler

But as the boiler is single-riveted we multiply the above quotient by .56, and we have the following:

$$\begin{array}{r} 650 \\ .56 \\ \hline 3900 \\ 3250 \\ \hline 364.00 \end{array}$$ lbs. Bursting pressure.

Q. Determine the pressure per square inch allowable on a new boiler to which the heat is applied on the outside of the shell, to be employed on the Mississippi River and its tributaries, which boiler is 40 inches in diameter, containing two flues 16 inches in diameter each, and shell made of homogeneous steel plates .26 of an inch thick, 60,000 pounds tensile strength, longitudinal seams double riveted and holes drilled? [Section 4434 Revised Statutes.]

A. That boiler would not be allowed for the reason that you can not get two 16 inch flues in a 40 inch boiler and have 3 inches space between and around the flues as required by Section 4434. This, of course, is a catch question, but any man who is well posted will not be caught by it. The thickness of material in flues is not given, neither is the length, and a candidate ought to detect that.

Q. Determine the pressure per square inch allowable for steel plates .375 of an inch thick, having a tensile strength of 70,000 pounds, intended for a new boiler to be employed on the Mississippi River and its tributaries, which boiler is required to be 46 inches in diameter, with longitudinal seams single-riveted and holes punched, and containing two flues each 16 inches in diameter? [Section 4434 Revised Statutes.]

A. The pressure allowable would be 190.21+ pounds if it were not for the law prohibiting the use of material over .30 of an inch thickness; and again, it will be seen that the thickness of the flues is not given, therefore it is impossible, without that information, to determine the pressure allowable, and no inspector would be allowed to determine the pressure except by taking the strength of the flues into consideration as well as the shell.

Q. Determine the pressure per square inch allowable on a boiler 48 inches in diameter, with plates .26 inch in thickness, and a tensile strength of 58,000 pounds per square inch, with holes drilled and longitudinal seams single riveted? [Section 4433, Revised Statutes, Rule 2, Section 3.]

A. This question is put direct except that part of it relating to the holes being drilled, but longitudinal seams are single riveted, that is designed to catch the candidate, as he may allow 20 per cent. on account of the holes being drilled. Therefore the question should be answered as follows:

```
        .26
       58000
      ------
      208000
      130
     -------
 24) 15080.00 (628.33+ and 628.33÷6=104.72+ lbs.   The pressure allowable.
     144
     ---
       68
       48
       --
       200
       192
       ---
         80
         72
         --
          80
          72
          --
```

Q. Determine the pressure per square inch allowable on a boiler 40 inches in diameter, with plates .24 of an inch thick, and a tensile strength of 60,000 pounds per square inch, with holes punched and longitudinal seams double-riveted? [Section 4433 Revised Statutes, Rule 2, Section 3.]

A. 120 pounds. No allowance is made on account of longitudinal seams being double riveted, because the holes are punched.

Q. Determine the pressure per square inch allowable on a required cylindrical boiler, 43 inches in diameter, longitudinal seams double riveted, holes drilled, boiler to be made of homogeneous steel plates, .2549″ in thickness, samples of which .2549″ x .98″ separated in testing, at the smallest point, under a strain of 14,996 pounds, and reduced in dimensions, at point of fracture to .89″ x .18″? [Section 4433 Revised Statutes, Rule 1, Sections 3 and 6; and Rule 2, Section 3.]

A. This is intended for a catch question, in which the candidate is required to demonstrate his ability to determine the tensile strength of boiler plate, ductility of boiler plate and the safe-working pressure, or rather the pressure allowable under the law. If he understands his business he will discover that the ductility of the material is but a fraction over 31 per cent., when Rule 1, Section 6, requires not less than 50 per cent., otherwise the boiler would be allowed a pressure according to the following operation:

$.2549 \times .98 = .249802$, area of sample. And breaking strain of sample (14,996 pounds) divided by area of cross section of sample, equals:

$$\left(\frac{(14996 \div .249802) \times .2549}{43 \times 2}\right) \times 1.2 = 142.34+ \text{ lbs.}$$

Q. Determine the hydrostatic pressure per square inch to be applied to a boiler 42 inches in diameter, with plates .26 of an inch in thickness, and a tensile strength of 65,000 pounds per square inch, longitudinal seams double riveted and holes drilled. Boilers to be employed on a passenger steamer? [Rule 2, Section 4.]

To determine the hydrostatic pressure the working pressure must first be obtained, and the whole to be performed as follows:

$$\left(\frac{65000 \times .26}{42 \times 2}\right) \times 1.2 = 160.94+ \text{ lbs. Working pressure.}$$

And one-half of the working pressure (80.47) added to the working pressure (160.94) equals 241.41 pounds, the hydrostatic pressure.

Q. Determine the steam pressure per square inch allowable (based on the diameter of stay bolts) on a fire-box boiler of the locomotive type with stay bolts in flat surface $\frac{15}{16}$ of an inch in diameter, and placed 5 inches apart from center to center. The bolts at the bottom of the thread being $\frac{3}{4}$ of an inch in diameter? [Rule 2, Section 6.]

A. RULE.—Multiply the area of the diameter of cross section of the stay bolt at the bottom of the thread by 6000, and divide the product by the number of square inches in the surface stayed by the bolt; the quotient will give the pressure allowable per square inch.

Example.—Let 75 one hundredths of an inch equal diameter of bolt at bottom of thread.

Let .7854 equal a constant.

Let 6000 equal a constant.

Let 5 inches equal distance from center to center of bolts.

Then we have:

$$\frac{.75\times.75\times.7854\times6000}{5\times5}=106.029 \text{ lbs. Pressure allowable.}$$

Q. Determine the required diameter of stay bolts placed 5 inches apart from center to center, to be allowed a pressure per square inch in the boiler of 150 pounds? [Rule 2, Section 6.]

A. Multiply the pressure per square inch by the number of square inches of surface to be stayed; then divide the product by 6000, and divide the quotient by .7854, and extract the square root of the last quotient; the answer will give the diameter of the bolt at bottom of thread required for a given pressure.

Taking the previous case, we have:

$$\sqrt{\frac{106.029\times25}{6000\times.7854}}=.75 \text{ Diameter of bolt at bottom of thread.}$$

Q. Determine the distance from center to center, stay bolts $\frac{3}{4}$ of an inch in diameter at the bottom of the thread, have to be placed in the flat surface of a boiler, to permit a pressure of 175 pounds per square inch to be carried in such boiler? [Rule 2, Section 6.]

A. To determine the distance from center to center of stay bolts, multiply the area of bolt at bottom of thread by 6000; then divide the product by the required pressure per square inch, and extract the square root of the quotient; the answer will give the distance in inches from center to center of stay bolt.

Taking the previous case, we have:

$$\sqrt{\frac{.75\times.75\times.7854\times6000}{106.029}}=5 \text{ inches. Distance from center to center of stay bolt.}$$

Q. Determine the width (for testing) of a sample of boiler plate having a thickness of .24 at the point intended for fracture? [Rule 1, Section 3.]

A. RULE.—Divide the area of one-fourth of a square inch by the given thickness, and the quotient will give the width required.

Example.—Let 25×100 equal one-fourth of a square inch.
Let .24 equal given thickness of material.

Then we have: $$\frac{100 \times .25}{.24} = 1.04+$$

Q. Determine the pressure per square inch allowable on a corrugated flue 38″ in diameter, with a thickness of material of one-half inch? [Rule 2, Section 14.]

A. 184.20+ pounds.

Q. Determine the pressure allowable on a cylindrical, lap-welded flue, 8 feet long, 38 inches in diameter, made of material one-half inch in thickness? [Rule 2, Section 15.]

A. 73+ pounds.

Q. Determine the thickness of material required for a cylindrical, lap-welded flue 6 feet long, 40 inches in diameter, to carry a steam pressure of 90 pounds per square inch? [Rule 2, Section 10.]

A. Multiply the diameter of the flue, in inches, by its length, in feet; then multiply the product by the pressure per square inch; then divide the last product by 89,600, and extract the square root of the quotient.

$$\sqrt{\frac{40 \times 6 \times 90}{89600}} = .49+ \text{ Thickness of material.}$$

Q. Determine the diameter of a lever safety-valve, for a boiler having a grate surface $4\frac{1}{2}$ feet in length and $4\frac{1}{2}$ feet in width. [Rule 2, Section 24.]

A. 3.59+ Diameter of valve.

Q. Determine the pressure per square inch required to raise a safety-valve of the following description:

Example.—Let 4 inches equal diameter of valve.
Let 9 pounds equal weight of vàlve and spindle.
Let 4 inches equal length of short arm of lever.
Let 20 inches equal distance of center of gravity of lever from fulcrum.
Let 40 inches equal length of long arm of lever.
Let 32 inches equal distance of center of weight from fulcrum.
Let 18 pounds equal weight of lever.
Let 150 pounds equal weight of weight.

A. 103.37+ pounds.

Q. Determine *diameter* of piston and length of stroke of fire pump required for passenger steamers. [Section 4471 Revised Statutes.]

A. Divide the number of cubic inches (100) required by the statute by the length of the stroke of the pump; then divide the quotient by .7854, and extract the square root of the last quotient; the answer will give the diameter of piston or plunger of pump required

Then we have:

$$\sqrt{\left(\frac{100}{8}\right) \div .7854} = 3.98+ \text{ inches.}$$ Diameter of plunger or piston.

· Length of stroke, 8 inches.

Q. Determine the diameter of steam drum legs for a battery of two cylindrical boilers, each boiler being 42 inches in diameter and 30 feet in length, containing two flues, each 15 inches in diameter, with the boilers so set that two-thirds of the outer surface of the shell is exposed to the heat of the furnace. [Section 4435 Revised Statutes.]

A. Divide the number of square feet of effective heating surface in one of the boilers by 2, and divide the quotient by .7854, and extract the square root of the last quotient; the answer will give the diameter of each leg of the steam drum.

The answer is obtained as follows:

	3.1416	Constant employed to obtain circumference.
	15	Diameter of flues.
	15 7080	
	31 416	
	47.1240	Circumference.
	2	Flues.
	94.2480	Circumference of two flues.
	3.1416	Constant employed to obtain circumference.
	42	Diameter of boiler.
	6 2832	
	125 664	
	131.9472	Circumference of boiler in inches.
	94.2480	Circumference of two flues in inches.
3)	226.1952	Total circumference of boiler and two flues in inches.
	75.3984	⅓ of circumference of boiler and flues in inches.
	2	
12)	150.7968	⅔ of circumference of boiler and flues in inches.
Am't carried forward,	12.5664	⅔ of circumference of boiler and flues in feet

Am't brought forward,	12.5664	⅔ of circumference of boiler and flues in feet.
	30	Length of boiler and flues.
	2) 376.9920	Number of square feet of heating surface.
	.7854) 188.4960	Area of steam drum leg.
	240	Square of diameter of leg.

The square root of 240 is obtained as follows:

240 (15.49+ inches. Diameter of steam drum leg. Answer to above question.
1

25) 140
125

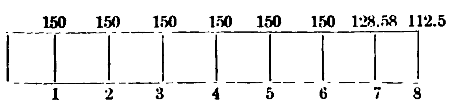
304) 1500
1216

3089) 28400
27801

Q. Determine the actual horse power of a pair of high pressure engines, on a steamer, with cylinders each 24 inches in diameter, 5 feet stroke, crank making 25 revolutions per minute, valve cutting off at three-quarters of the stroke, initial pressure 150 pounds, clearance not taken into consideration. [Section 4415 Revised Statutes.]

A. Multiply the area of the diameter of the cylinder by the average pressure per square inch throughout the stroke (after deducting loss due to friction); then multiply the product by the number of feet the piston travels per minute; then divide the last product by 33,000; the answer will give the horse power of one engine. In this case there are two engines; therefore the horse power of one must be multiplied by two; and in this case the answer is 591.62+ horse power.

Example.—



$(150 \div 7) \times 6 = 128.58$

$(150 \div 8) \times 6 = 112.5$

150
150
150
150
150
150
128.58
112.5

8) 1141.08

142.635 lbs. Average pressure throughout the stroke.

Let friction equal $2.5 + (\overline{142.635 + 14.7 - 2.5}) \times .075 = 14.112+$

Then we have:

$$\frac{24\times24\times.7854\times(142.635-14.112)\times5\times2\times25}{33000}=440.47+ \text{ horse power.}$$

And $440.47\times2=880.94$ horse power of two engines.

Q. Determine the nominal horse power of a pair of high pressure engines of the same dimensions as those above? [Section 4415.]

A. Multiply the square of the diameter of the cylinder by the velocity (250 feet per minute), and divide the product by 1000, the quotient will give the nominal horse power of the engine.

EXPLANATION.

The rule for determining the nominal horse power of an engine is based on an average pressure throughout the stroke of 60 pounds per square inch, and a velocity of piston of 250 feet per minute.

Therefore, taking the pair of engines referred to above, we have:

$$\left(\frac{24^2\times250}{1000}\right)\times2=288 \text{ Horse power for both engines, or 144 horse power each.}$$

Q. Determine the nominal horse power of a cylindrical horizontal flue boiler, 24 feet in length, 48 inches in diameter, containing two flues, each flue having a diameter of 16 inches; two-thirds of the shell being exposed to the heat of the furnace, and one-half of each flue being effective heating surface? [Section 4415 Revised Statutes.]

A. 25.13+ nominal horse power. This answer is based on 12 square feet of effective heating surface to the horse power in the boiler.

Q. Determine the bursting pressure of a steam engine cylinder having an inside diameter of 24 inches, a thickness of metal of $1\frac{1}{4}$ inches, and a tensile strength of 25,000 pounds per square inch? [Section 4415 Revised Statutes.]

A. 2604.16+ pounds.

Example.—
$$\frac{25000\times1.25}{24\div2}=2604.16+ \text{ lbs. The bursting pressure per square inch.}$$

Q. How much lap must a slide valve have with a travel of 4 inches and a lead of $\frac{1}{16}$ of an inch of an engine having a length of stroke of 5 feet, required to cut off when three-fourths of the stroke is reached? [Section 4415 Revised Statutes.]

A. Deduct from the entire length of the stroke of the engine the length of the stroke that is to be made before the steam is cut off; then divide the remainder by the full stroke, and extract the square root of the quotient, and multiply this root by half the entire travel of the valve, and from this travel subtract half the lead; the remainder will

give the lap required to cut off steam at any given point in the stroke of the engine.

Then we have:

$$\sqrt{\frac{5-3.75}{5}}=.5 \text{ and } .5\times(4\div2)=1, \text{ and } 1-\left(\frac{.0625}{2}\right)=.96875 \quad \text{Lap required to cut off at three-quarter stroke.}$$

Q. A steam pipe having an inside diameter of 4 inches, being supplied with a flat-seated valve 5 inches in diameter, what must be the amount of lift of the valve to produce an area of opening equal to the area of cross section of the opening of the steam pipe? [Section 4415 Revised Statutes.]

A. Divide the area of the pipe by the circumference of the valve, the quotient will give the required lift of the valve.

$$\frac{4^2\times.7854}{5\times3.1416}=.8 \text{ of an inch.} \quad \text{Lift required}$$

Q. Determine the thickness of material required for a cylindrical boiler having a diameter 44 inches, longitudinal seams single riveted, tensile strength of material 60,000 pounds per square inch, and to carry a pressure of 113.63 pounds per square inch? [Section 4415 Revised Statutes.]

A. Multiply the given pressure by the radius of the diameter of the boiler, then multiply the product by 6, and divide the last product by the given tensile strength of material; the quotient will give the thickness required in hundredths of an inch for single-riveted longitudinal seams.

Then we have:

$$\left\{113.63\times\left(\frac{44}{2}\right)\times6\right\}\div60000=.2499+ \text{ of an inch.} \quad \text{Thickness of material required.}$$

Q. Determine the thickness of material required for a cylindrical boiler, 40 inches in diameter, made of material having a tensile strength of 65,000 pounds per square inch, holes drilled and longitudinal seams double riveted, and required to carry a steam pressure of 150 pounds per square inch? [Section 4415 Revised Statutes.]

A. Multiply the given pressure by the radius of the given diameter of boiler, and multiply the product by 6; then divide the last product by the given tensile strength, and multiply the quotient by .83⅓; the answer will give the thickness in hundredths of an inch.

$$\frac{150\times(40\div2)\times6}{65000}\times.83\tfrac{1}{3}=.2307+ \text{ of an inch.} \quad \text{Thickness of material required.}$$

Q. Determine the tensile strength required for a boiler 42 inches in diameter, made of plates .26 of an inch in thickness, longitudinal seams single-riveted, and to carry 125 pounds to the square inch? [Section 4415 Revised Statutes.]

A. Multiply the given pressure by 6, and multiply the product by the radius of the diameter of the boiler, then divide the product by the thickness of the material; the quotient will give the tensile strength required.

$$\frac{125\times6\times(42\div2)}{.26}=60576+\text{ lbs.}$$

Q. Determine the diameter of a boiler required to carry a steam pressure of 150 pounds per square inch, made of material .25 of an inch thick, double riveted, holes drilled and tensile strength 60,000 pounds per square inch? [Section 4415 Revised Statutes.]

A. Multiply the thickness of the material by the tensile strength and call this "Product No. 1;" then multiply the given pressure by 6 and call this "Product No. 2;" then divide "Product No. 1" by "Product No. 2," and multiply the quotient by 2; add 20 per cent. to the last product, and the answer will give the diameter of the boiler.

Then we have:

$$\left(\frac{.25\times60000}{150\times6}\times2\right)+20\text{ per cent.}=40\text{ inches. Diameter of boiler.}$$

CHAPTER XVII.

A UNITED STATES BATTLE SHIP.

Figs. 194 and 195 represent the general arrangement of machinery of a triple-screw, protected cruiser of 7350 tons.

The boilers and machinery of this ship are designed to propel the vessel at sea at a maintained speed of 21 knots per hour. To attain that speed her motive machinery has been constructed so as to be capable of developing from 20,000 to 21,000 indicated horse power.

The power of this vessel has been divided into three parts, each being developed by a separate, triple-expansion engine, driving its own screw. In case of accident to any of her engines, this cruiser would still have a reserve power sufficient to drive her at a good rate of speed. In fact, it would be almost impossible to imagine or conceive a combination of circumstances or accidents that would render her entirely helpless.

A leading feature of the design is, that by means of a clutch coupling, either propeller can be disconnected from its engine and left free to revolve, thus retarding the way of the ship but very slightly, when she is being propelled by one or two engines.

A still further advantage is, that in moderate cruising, say with one-third power, a few boilers can be used with the high steam pressure for which they are intended, and one engine driven at the full power for which it is designed; by this means the power will be obtained economically, instead of wastefully, as it is when a large engine is running with low power. It is estimated that with one-third power and one screw, the ship can be driven about 15 knots per hour; with two screws and two-thirds power, from 18 to 19 knots per hour—the screws not in use being allowed to revolve freely in either case.

The foregoing are the reasons for adopting the plan of three screws instead of two, and not because three screws possess any other advantage over two. On the contrary, it was supposed by the designers, that so far as speed is concerned, that two screws would have a slight advantage over three, by the application of the same power.

Experiments have demonstrated that when three propellers are placed abreast of each other, the efficiency of the center one is greatly impaired by the interference of the side ones by the water flowing to

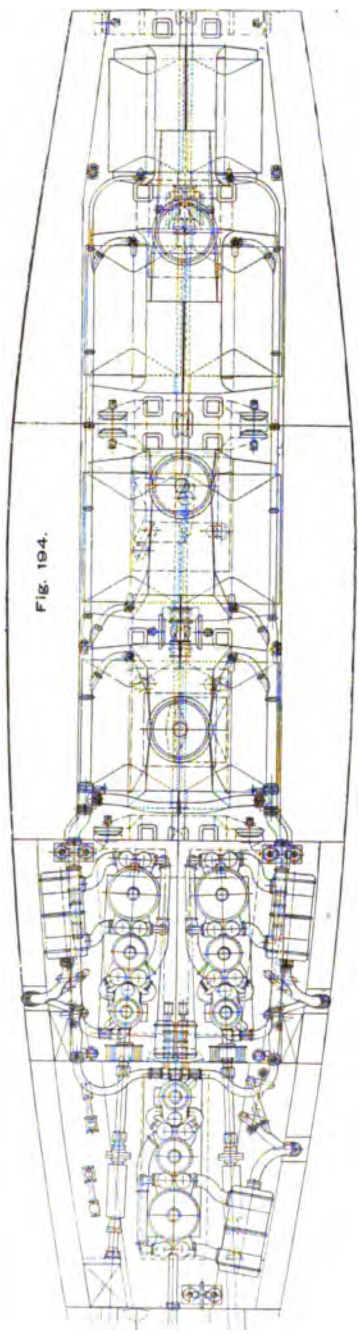


Fig. 194.

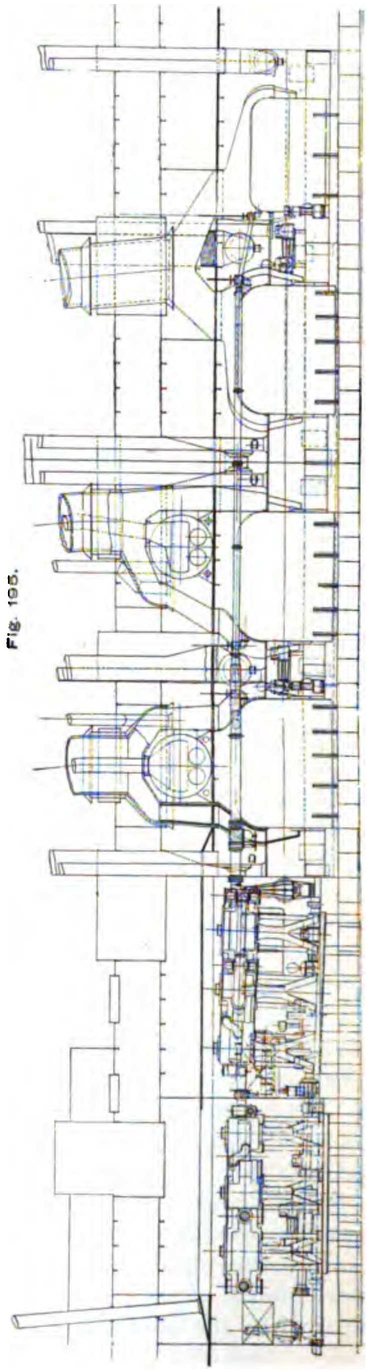


Fig. 195.

General Arrangement of Machinery.

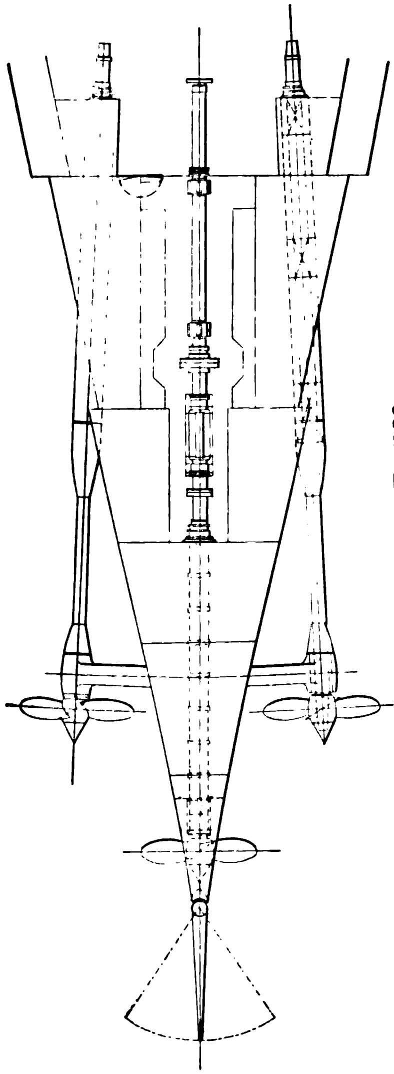
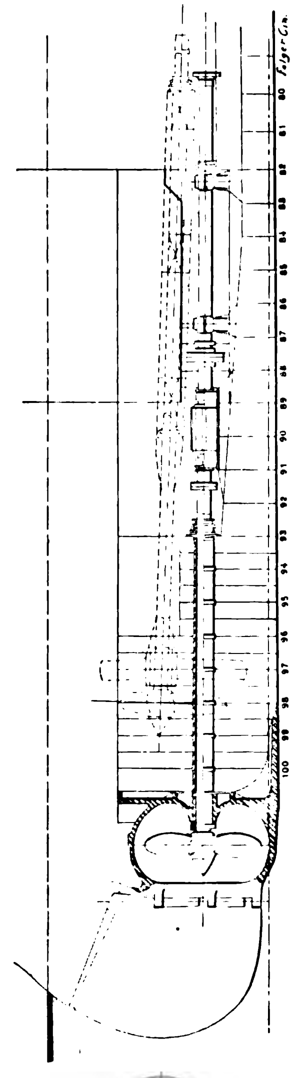
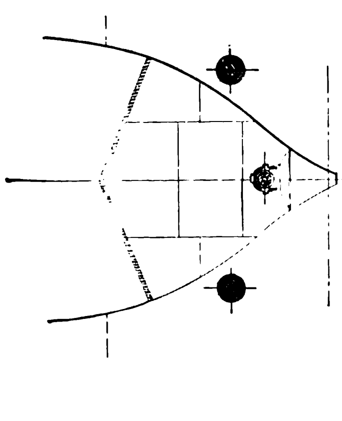


Fig. 196



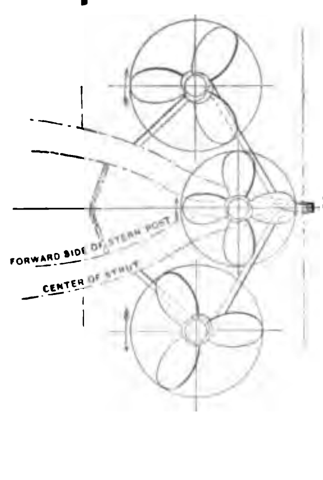
SECTION THROUGH CENTER SHAFT.

Fig. 197



LOOKING AFT.

Fig. 198



LOOKING FORWARD.

Fig. 199

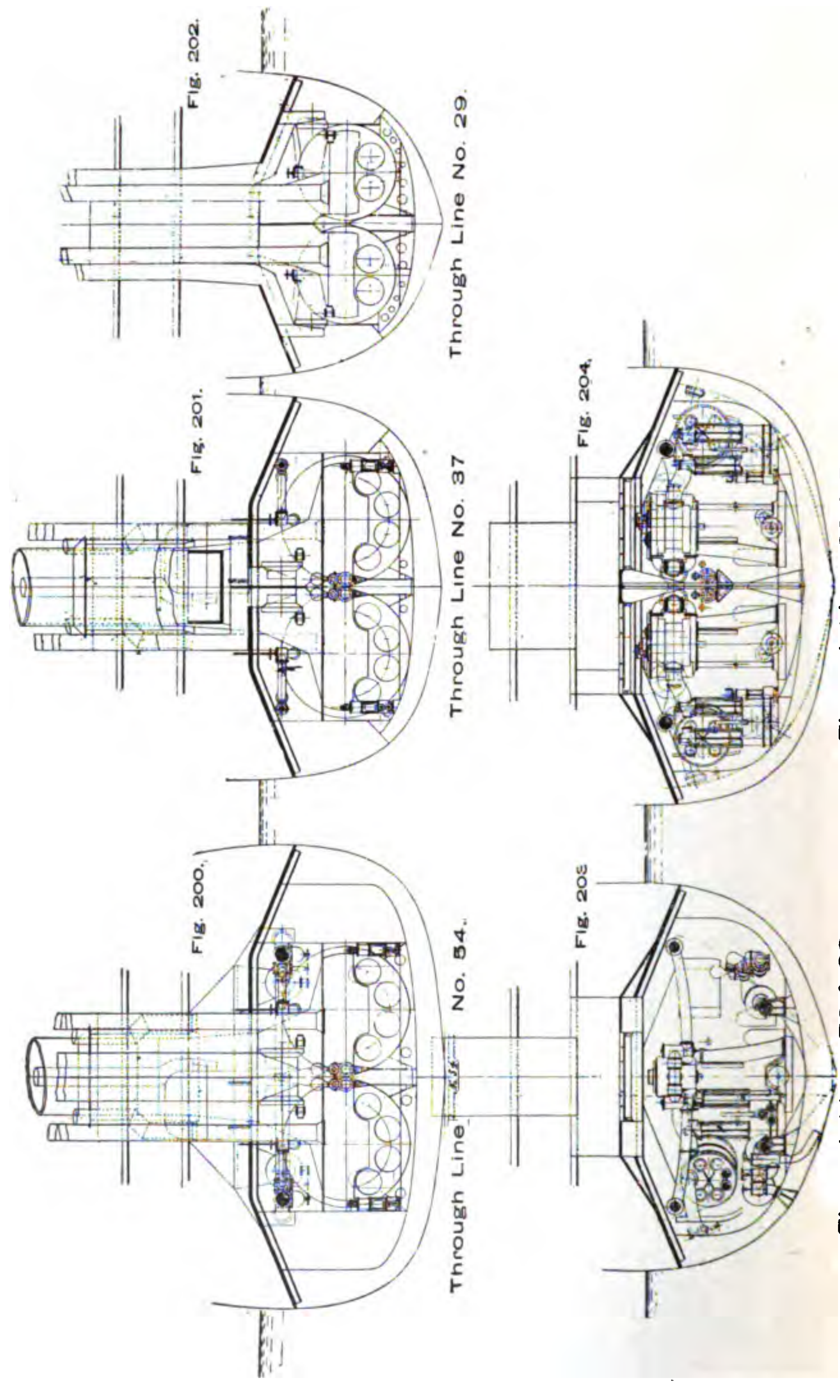


Fig. 202. Through Line No. 29.

Fig. 201. Through Line No. 37.

Fig. 200. Through Line No. 54.

Fig. 204. Through Lines 63 & 72.

Fig. 203. Through Lines 72 & 82.

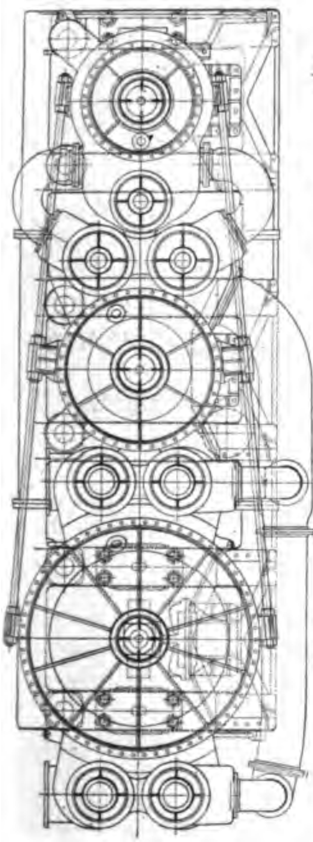


Fig. 205.

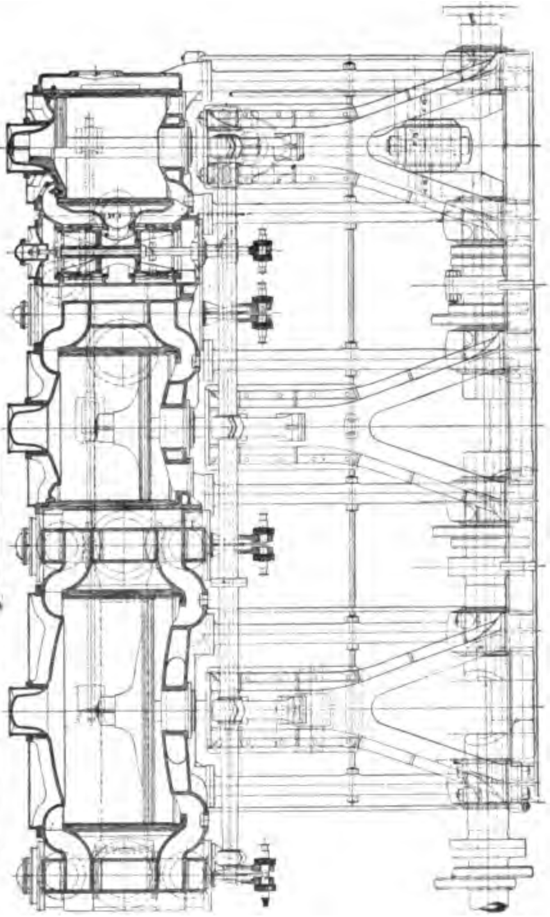


Fig. 206.

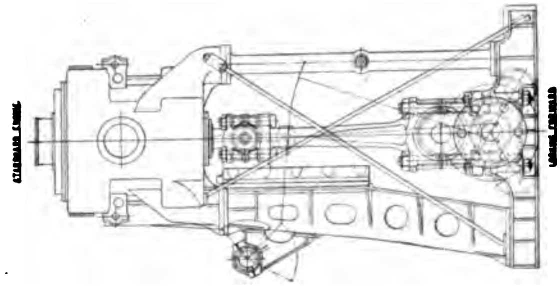


Fig. 207.

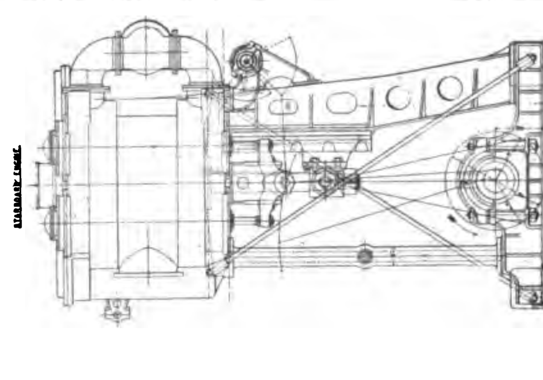


Fig. 208.

it. For this reason, the side propellers have been placed in this cruiser about fifteen feet forward of the center one, thus avoiding, as much as possible, working the latter in the race of the former. It will also be noticed, that in looking at the ship from aft (Fig. 199), the propellers are not in the same horizontal line, but that the side ones are placed as high as their diameters will permit; also, that to bring these propellers clear of the ship's sides, the side shafts incline outwards from the center line of the ship 4 degrees, as shown in Fig. 196, and they also incline slightly upwards. The shaft of the center propeller is in the central longitudinal plane of the ship, and inclined slightly downwards.

The central screw has four blades, and about ten per cent. more pitch than the side ones, as it works in more or less disturbed water. The side screws are three bladed, and the blades of all the propellers are adjustable, and are set to the pitch, which, upon trial, is found to be most efficient.

MAIN ENGINES.

The main engines have cylinders of 42, 59 and 92 inches diameter, with a common stroke of 42 inches. They are arranged in three water-tight compartments, each being complete in itself with all of its auxiliaries, and entirely independent of the others, so that one or two might be completely disabled without in the least interfering with the working of the other.

Figs. 205, 206, 207 and 208 show the plan, sectional elevation and end views of one of the set of triple-expansion engines.

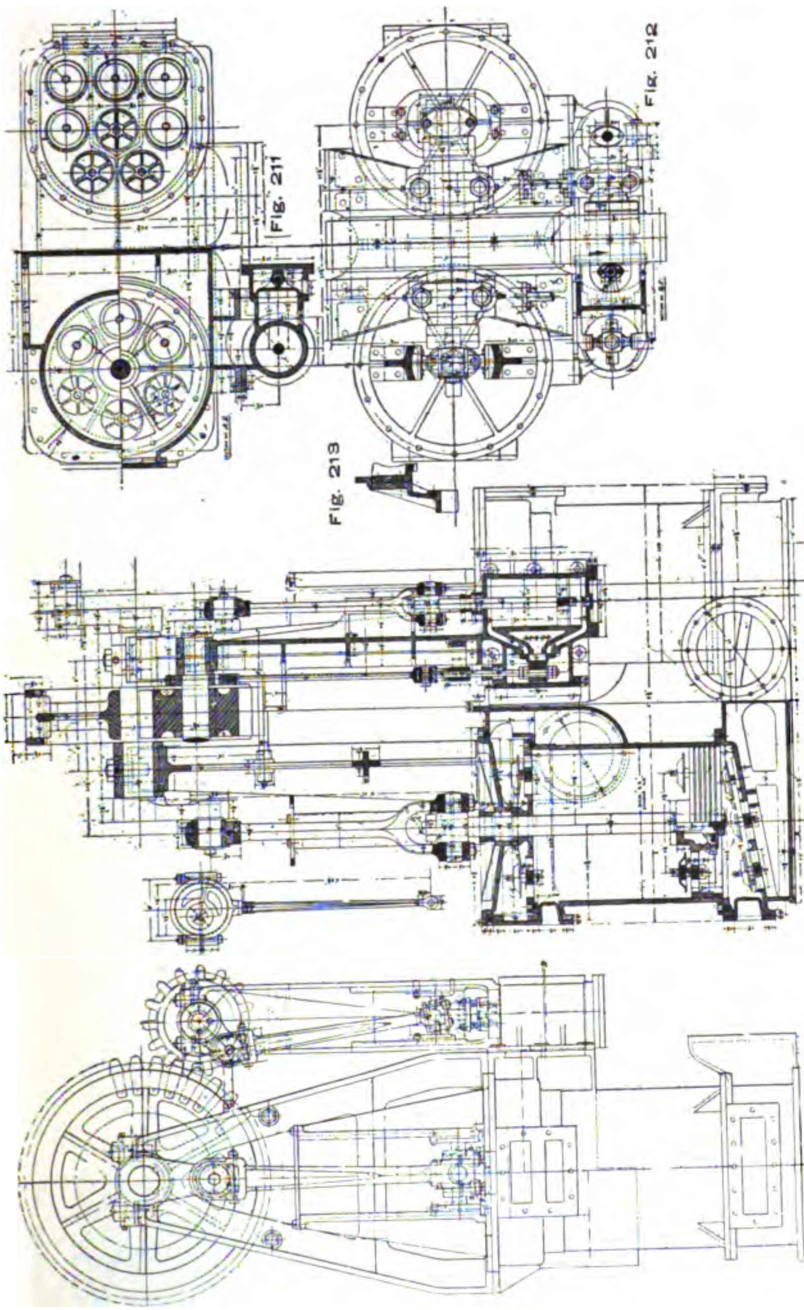
AIR PUMPS AND CONDENSERS.

The air pumps are two vertical, single-acting, lifting pumps for each engine, the pumps being 22 inches in diameter by 20 inch stroke. They are driven by two simple engines, with cylinders 7 inches in diameter and 12 inch stroke, arranged to exhaust into the condenser or into either receiver. Each engine is geared to make two and one-half revolutions for each double stroke of the pump to which it is attached.

There are three condensers of the type shown in the engravings, with a cooling surface in each of 9474 square feet.

MAIN AND AUXILIARY BOILERS.

Steam is supplied by eight main, and two auxiliary boilers, all built of milled steel. Six of the main boilers are each 15 feet 6 inches in diameter outside, and 21 feet 3 inches in length; and two are each 11 feet 8 inches in diameter, and 18 feet 8½ inches in length. The shells of the large boilers are $1\frac{21}{64}$ inches in thickness; those of the



Air-Pumps and Engines.

Fig. 209. Fig. 210. Fig. 211. Fig. 212. Fig. 213.

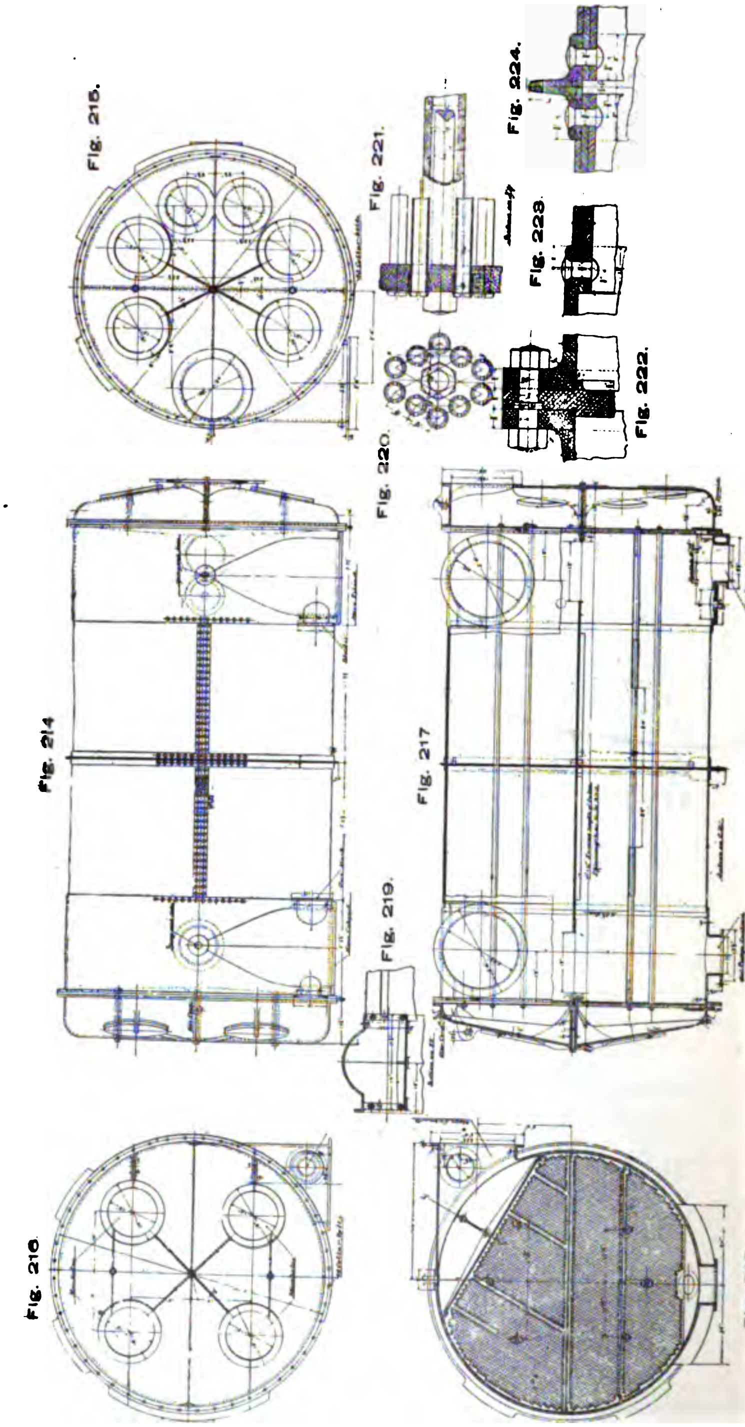


Fig. 215.
Fig. 221.
Fig. 224.
Fig. 223.
Fig. 222.
Fig. 220.
Fig. 214
Fig. 217
Fig. 219.
Fig. 216
Fig. 218

Condenser.

Diameter of Boiler Outside,	15 ft. 6 in.
Diameter of Furnace Inside,	3 ft. 3 in.
Length of Grate,	6 ft. 9 in.

HEATING SURFACE.

Tubes,	5177.00	sq. ft.
Furnaces,	375.40	" "
Comb. Chambers,	380.00	" "
Total,	5932.4	" "

Main Boiler—Six of This Kind.

Grate Surface,			175.5 sq ft
Ratio of Heating to Grate Surface,			33.8
Calorimeter,			22.2
Ratio of Grate Surface to Calorimeter,			7.9
Boiler Pressure,			160 lbs
No. of Tubes,	1128	312 stay	6 B W G
		816 ord'n'y	12 B W G.

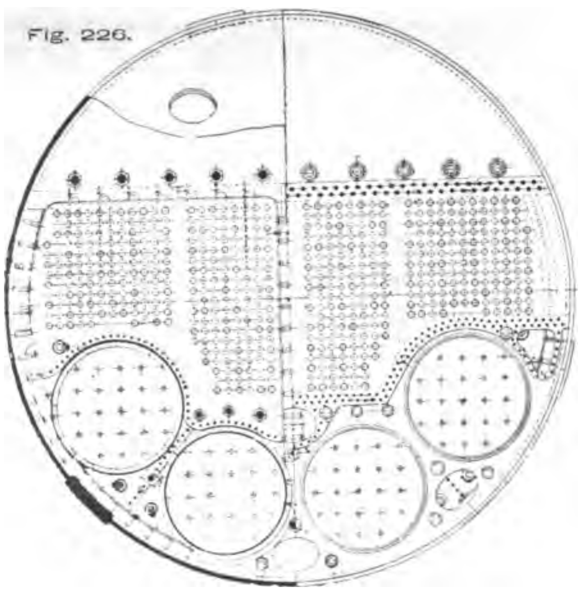


Fig. 226.

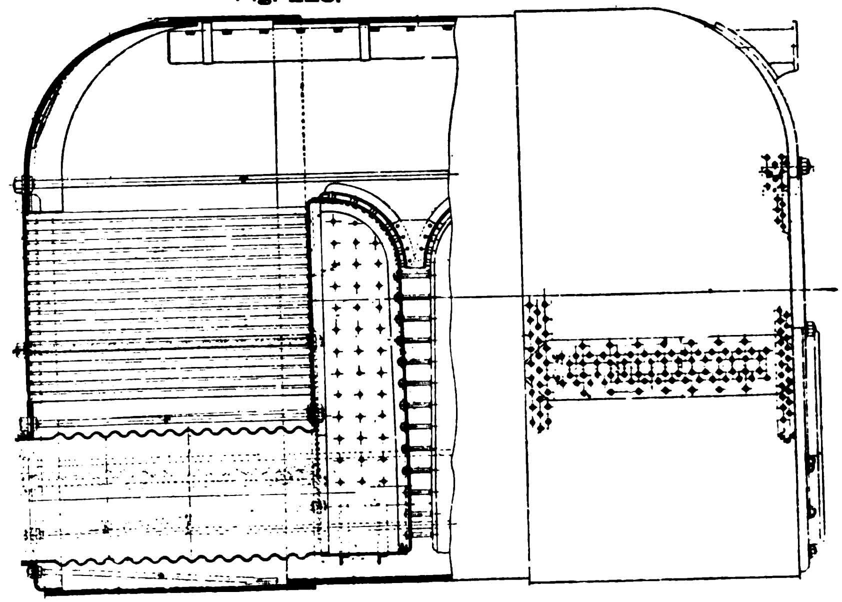


Fig. 225.

Main Boiler

Two of This Kind.

Diameter of Boiler Outside	11 ft. 8 in.
Diameter of Furnace Inside	3 ft. 6 in.
Length of Grate	6 ft. 0 in.

HEATING SURFACE.

Tubes	2497.0 sq. ft.
Furnaces	171.2 " "
Comb. Chambers	202.0 " "
Total	2870.2 " "

Grate Surface,	84 sq. ft.
Ratio of Heating to Grate Surface	34.2
No. of Tubes	632 { 156 stay 6 B. W. G. 476 ord'n'y 12 B. W. G.
Calorimeter,	6.47 for one end.
Ratio of Grate Surface to Calorimeter	6.5
Boiler Pressure,	160 lbs.

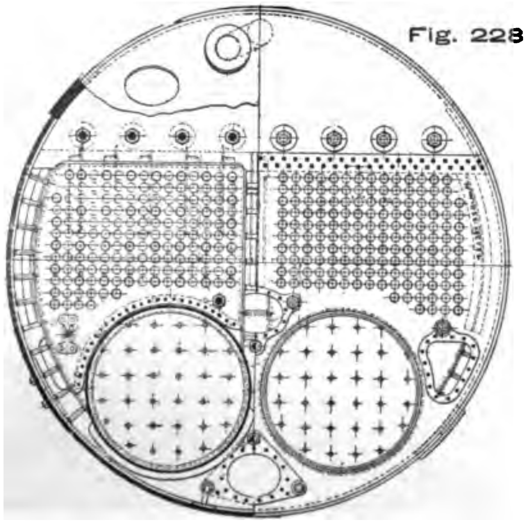


Fig. 228

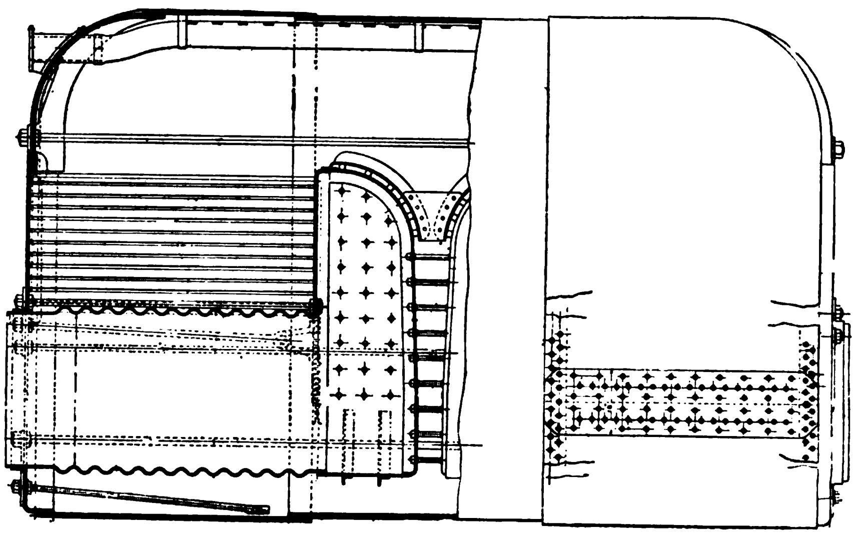


Fig. 227.

Auxiliary Boiler.

Two of This Kind.

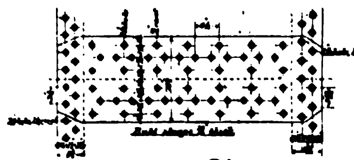


Fig. 231

Diameter of Boiler Outside,	10 ft. 1 [illegible] in.	
Diameter of Furnace Inside,	2 ft. 9 in.	
Length of Grate,	5 ft. 10 in.	

HEATING SURFACE.

Tubes,	824.66	sq. ft.
Furnaces,	60.0	" "
Comb. Chamber,	84.0	" "
Total,	968.66	" "

Grate Surface,	32 sq. ft., one boiler.
Ratio of Heating to Grate Surface,	30.3
Calorimeter,	5.9
Ratio of Grate Surface to Calorimeter,	5.4
Boiler Pressure,	160 lbs
No. of Tubes,	216 { 58 stay 6 B. W. G; 158 ord'n'y 12 B. W. G

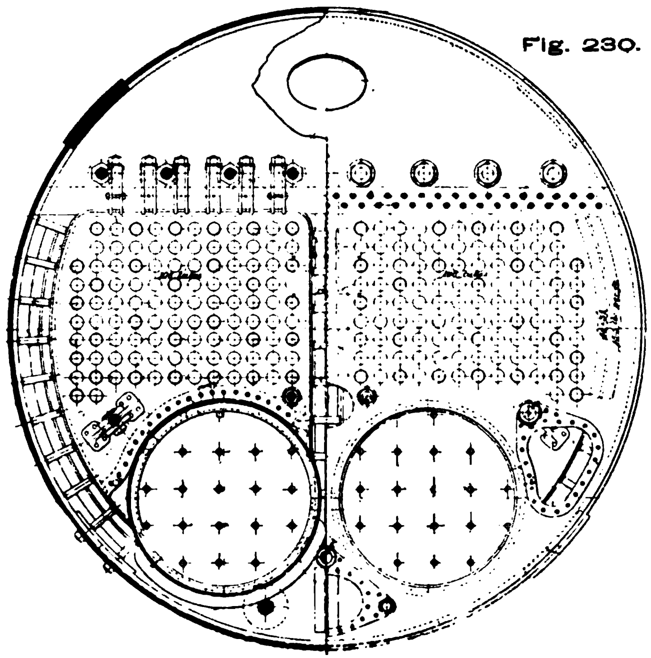


Fig. 230.

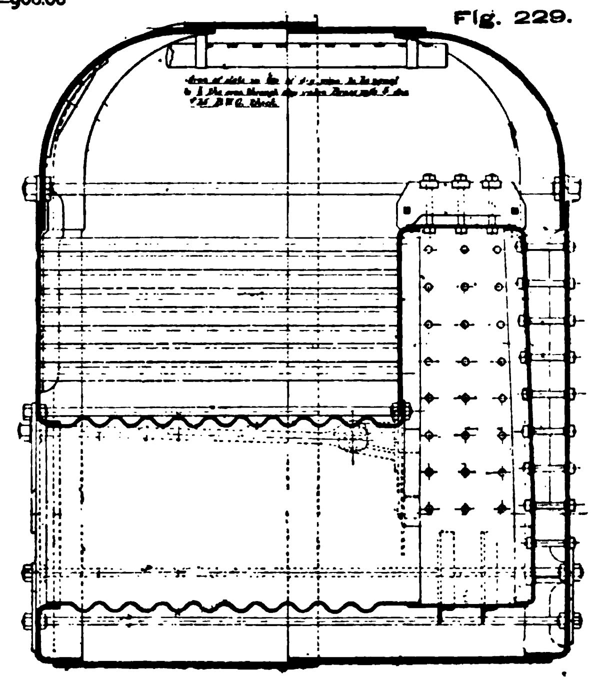


Fig. 229.

smaller are $1\frac{1}{64}$ inches in thickness. The auxiliaries are each 10 feet $1\frac{25}{32}$ inches outside diameter, and 8 feet 6 inches in length, and the shells are $\frac{57}{64}$ of an inch thick. All of the boilers are constructed to carry a safe-working pressure of 160 pounds per square inch.

All of the main boilers are of the Scotch type, double ended, and with $2\frac{1}{4}$ inch steel tubes. Each large boiler has eight corrugated steel furnaces of 3 feet 3 inches inside diameter; and each small boiler has four such furnaces 3 feet 6 inches inside diameter.

The heating surface of each of the large boilers is 5932 square feet, and of each small boiler 2870 square feet, and of each auxiliary boiler $968\frac{1}{2}$ square feet, making a total of 43,269 square feet.

The total grate surface is 1285 square feet—each large boiler having $175\frac{1}{2}$ square feet, and each small boiler 84 square feet, and each auxiliary boiler 32 square feet. The auxiliary boilers are placed above the protected deck, as shown in Fig. 195.

The main boilers are arranged in groups of two, in four separate water-tight compartments, with five athwartship fire rooms and three smoke pipes.

The forced draught is on the closed fire room system; each fire room being supplied with centrifugal fan blowers.

The feed water is supplied by a main and auxiliary feed pump in each working fire room.

There are evaporators to make up the loss of fresh water to the boilers, and to supply the distillers, reversing and turning engines, fire and bilge pumps, ice machines, steam ash hoists, capstan engine, and all other auxiliaries fitted in the most modern ships.

CHAPTER XVIII.

WESTERN RIVER STEAMBOAT PRACTICE.

SPECIFICATIONS FOR ENGINES AND BOILERS FOR A WESTERN RIVER STERN-WHEEL STEAMER.

The following are specifications for engines and boilers for two high-pressure puppet-valve lever engines, with adjustable cut off, to be regulated by suitable levers and lines, and requiring but one cam rod each.

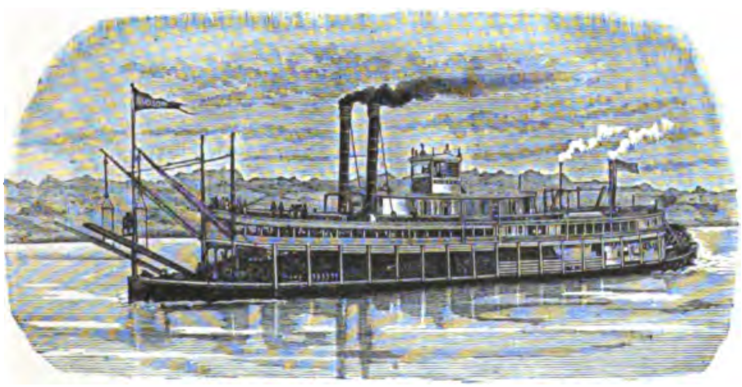


Fig. 232.
STEAMER HUDSON.

SIZE.

Diameter of cylinders, 12 inches; stroke, 5 feet; side pipes provided with receiving valves 3¼ inches in diameter, and exhaust valves 4¼ inches in diameter, to be neatly planed, the caps turned and faced on both sides, and connected to side pipes by a Calvin joint.

CROSS SHAFTS, CYLINDERS, PISTONS, VALVES, ETC.

Cross shafts of wrought-iron, independent of side pipes, to be neatly planed, provided with quarter brasses, and the journals bored out and bolted to side pipes.

Puppet heads to be turned and steel valve stems fitted in taper, moistened, and provided with steel keys and rings; the levers to be neatly finished and air cushions placed under them; connection

between side pipe and cylinder to be made with copper joint; piston rods of steel, 2¾ inches in diameter, with heads suitable for four elliptic springs, and provided with cast-iron spring ring; brass spring packing, cast with clips and recesses, the latter to be filled with best babbitt. The packing and follower to be hand scraped; the heads faced and ground on head and follower; the studs supporting springs to be turned and chased and provided with two finished brass nuts each; piston-head bolts of good iron, faced nuts, and copper nut between nut and follower; tee heads of cast-iron, with wrist no less than 3½ inches in diameter, and each web 4 inches deep by 2 inches wide, fitted with top, bottom, and quarter brasses, adjustable by set screws, and so arranged that they may be removed or replaced without unshipping the tee head. Cylinders will be fitted with Hammer's automatic cylinder cocks.

Bed plates to be of form embodying the most strength with the least weight; the bearings to be planed to meet the planed surfaces of cylinder lugs, provided with lug key keepers and keying plates at each end.

SLIDES.

Slides of cast-iron, free from all imperfections, will be truly and evenly planed; will be supported on iron bed plates, and so arranged as to be adjustable with set screws held in place by jam nuts.

REVERSING GEAR.

The reversing gear will consist of wrought-iron fork on full-stroke rod, with proper lever and rod connections, with a wrought-iron cross shaft, secured to deck by cast-iron journal blocks, and provided with lever and counterpoise near throttle for operating the same.

PILLOW BLOCKS.

The pillow blocks, of cast-iron, will be of form suitable for steel cylinder beams, of size sufficient to admit of a journal 7 inches; the journal should be not less than 8 inches long; the blocks to be planed on both sides, fitted with bottom and quarter brasses, of best copper and tin, and accurately and truly bored out.

Cam-yoke brackets are independent of blocks, and are held in recesses cast in blocks by bolts passing down to cylinder beams; the bottom of block will be planed to a close fit with the beams, and retained in place by wrought or cast-iron keying plates riveted to the cylinder beams.

PITMANS.

Pitmans to be made of wrought-iron, 22 feet from center to center, to be forged solid on ends and cut out to receive brasses, and have

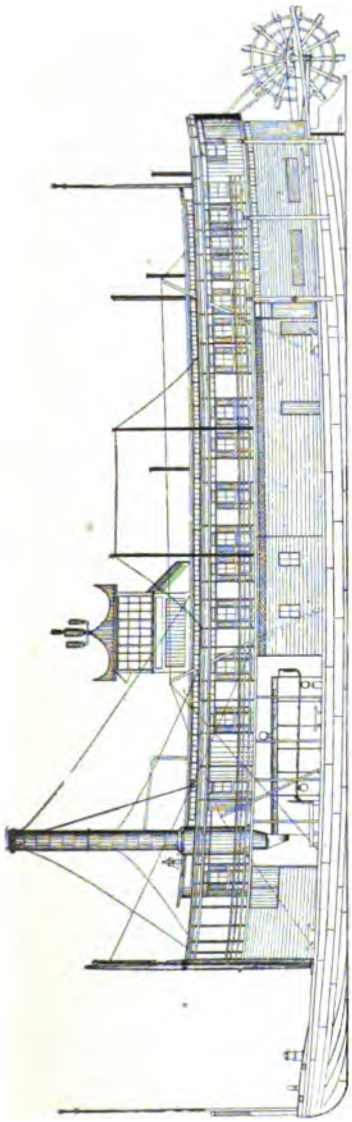


FIG. 233 Elevation.

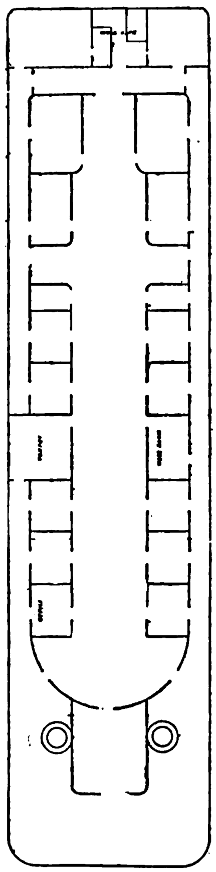


FIG. 234. Plan of Cabin Deck.

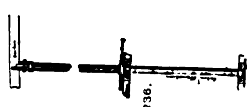


FIG 236.

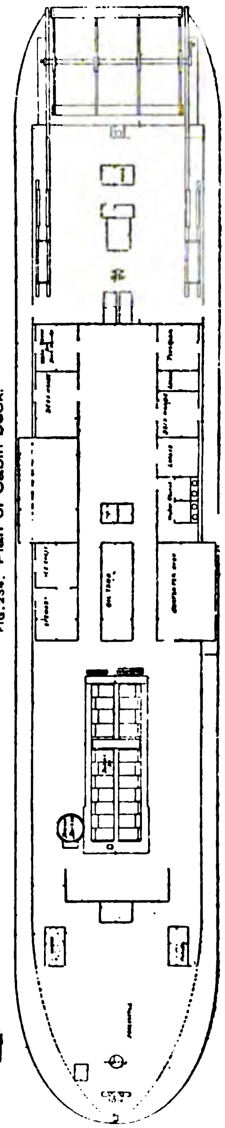


FIG. 235. Plan of Main Deck.

open ends similar to wooden pitman; the body of pitman to be 3⅛ inches, round at ends, and tapered to within 1½ inches of center to 4¼ inches diameter, where boss, 5½ inches diameter, 3 inches long, to receive center studs and stiffening rods of 1 inch on top and bottom, secured to both ends; pitmans to be neatly turned, planed, and finished, and fitted with good copper and tin brasses, to be truly planed and bored; crank wrist drilled and babbitted; the gibs and keys to be planed, and the key held in place by a key attached with jaw to gib, and passed through lug on head of key, and held in place by a nut above and below.

THROTTLE.

The throttle chamber will have the main valve in center, the doctor and blow-off valve on either side, and the main and branch steam-pipe chambers will be in line across the throttle; the chambers will be evenly planed, the caps turned and finished, and the connections Calvin joints; valve stems and screws of steel, and swivel of wrought-iron.

SHAFT.

Shaft of best hammered new bar iron, not less than 7. inches diameter in journals, with hexagon body, 7 inches across the square; four flanges 36 inches in diameter, 14 inches in sub-socket, 11 inches in socket, 2 inches deep, and to have 12 arms, and weigh not less than 400 pounds. Two wrought-iron journal and cam collars will be placed on the shaft, and the flanges secured by iron wedges.

CRANKS.

Two wrought-iron cranks, to be fairly planed and the holes accurately bored to face. The wrought-iron wrists, not less than 3½ inches in diameter, will be fitted in cranks, with a taper of three-quarter inch to the foot.

PUMPS.

One double-acting deck pump, with chambers to contain not less than 100 cubic inches of water each, and connected with a suitable pipe passing through the side of the vessel so low as to be at all times under water when the vessel is afloat. And there shall also be a pump of sufficient strength and capacity, and suitably arranged, for testing the boilers.

DOCTOR.

One good, substantial doctor, with area of cylinder sufficient to work doctor with 50 pounds of steam to the square inch; spring packing in the cylinder, steel piston rod, and gun metal slide valve; the bases for columns and pump chambers on bed plate to be accurately

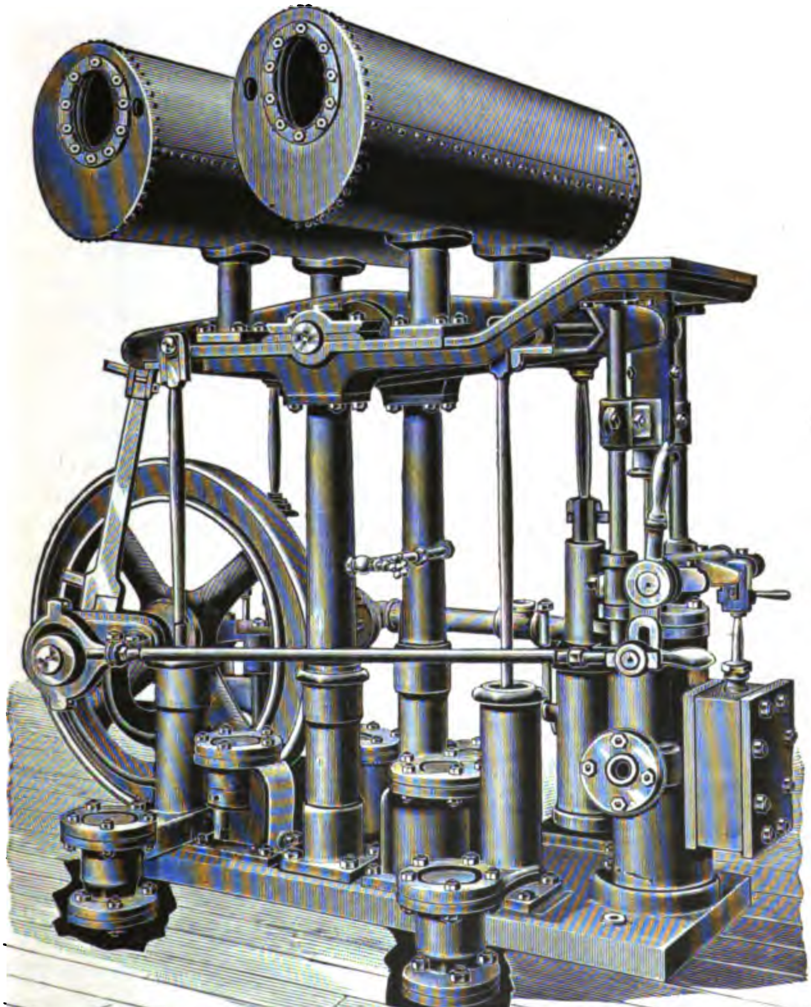


Fig. 237.

WESTERN RIVER STEAMBOAT DOCTOR.

planed, so that all parts will come squarely together; hot water pumps to have chambers bored, and fitted with copper and tin valves and seats, the seats to be driven in; all valve seats to be securely fastened, either by set screws or by riveting over at bottom; the discharge valve chamber to stand above the opening from the pumps; caps on pumps (round) to be finished and the joints of thin lead, each cap to have four bolts with finished nuts; hot water pump plungers, $3\frac{1}{2}$ inches diameter by 10 inch stroke, or their equivalent, of good, hard iron, free from flaws, and fitted with spade handle and adjustable strap; all square brasses in strap and joints to have set screw keepers.

Cold water pumps to be of sufficient capacity to supply water to ash pan and piston rods, in addition to an ample supply to the hot water pumps; caps similar to those of hot water pumps; walking beam of cast-iron, banded with wrought-iron; wrought-iron shaft and wrists for pump rods, and good, heavy fly wheel.

HEATERS.

Two heaters, 22 inches in diameter by 5 feet long; copper, hard rolled, 70 pounds to the sheet, with a copper worm in each, 18 feet long, $2\frac{1}{2}$ inches diameter, of 45 pound sheets, to be well secured in heater. A wrought-iron floor, of No. 10 gauge, will be placed in each heater above the head of water, to prevent the exhaust from throwing water from the heaters.

NIGGER BOILER.

There must also be furnished an auxiliary boiler, with necessary pipes and attachments, and a No. 4 donkey steam pump, with the necessary pipe connections, fire apparatus, etc., and a water cylinder, 3 inches in diameter and 8 inch stroke. Pipes from donkey to fire apparatus to be finished copper.

The boiler to be 42 inches in diameter, 7 feet 6 inches high, of the vertical submerged tubular style; fire box, 26 inches high, 35 inches diameter; cone, 34 x 19 inches in diameter, 18 inches high; the boiler to contain forty-four 3 inch lap-welded tubes, $33\frac{1}{2}$ inches long, with ash pan, grates, steam and water gauges, check and blow-off valves; steam upright fly wheel, with pump to supply the same; the boiler to be supplied with a spring-loaded safety-valve, $1\frac{9}{10}$ inches in diameter, of the style and construction approved by the Board of Supervising Inspectors of Steam Vessels, and to conform in all respects to the requirements of the United States laws. The shell of the boiler to be made of homogeneous steel $\frac{30}{100}$ of an inch thick, with a tensile strength per square inch of not less than 60,000 pounds, and to show a contraction of area of not less than fifty per cent. at point of fracture in a test of the material; and longitudinal seams to be double riveted, with the best quality of rivets; the fire box to be of such thickness of

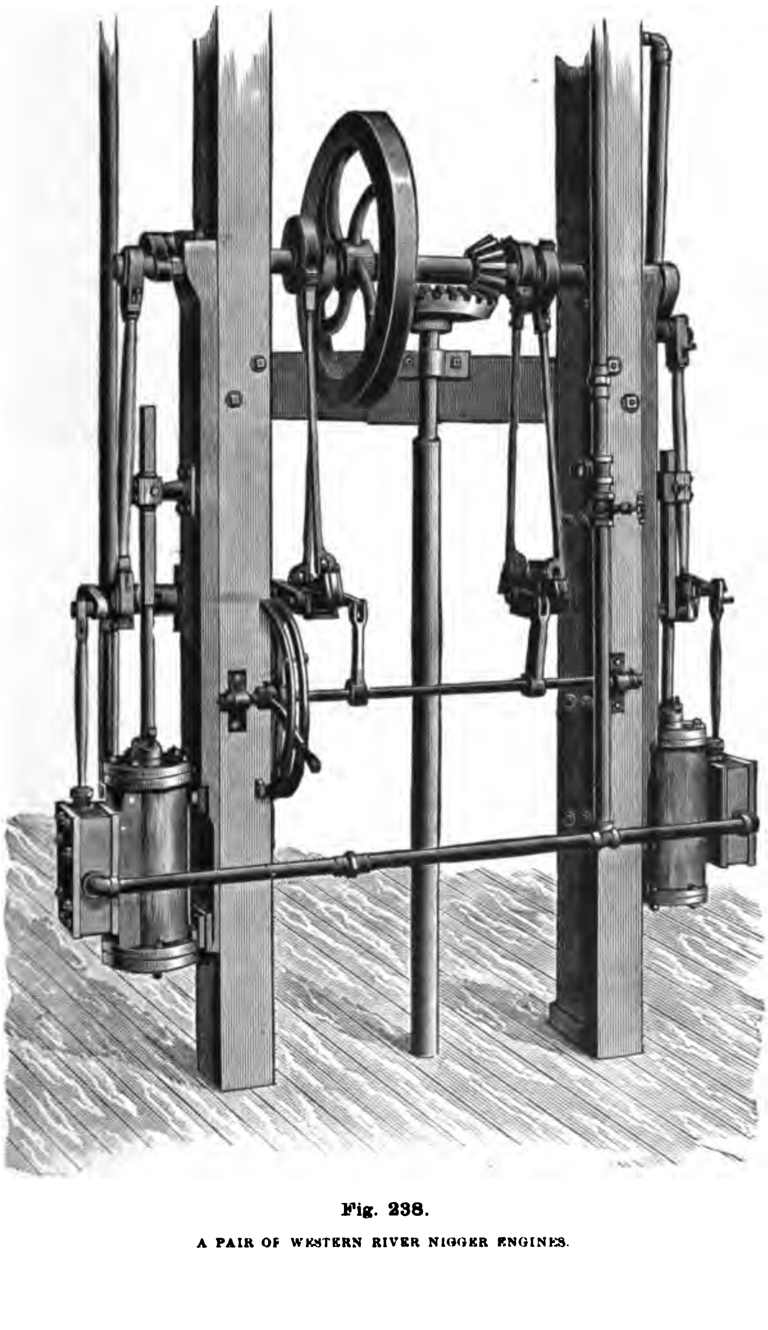


Fig. 238.

A PAIR OF WESTERN RIVER NIGGER ENGINES.

material, and all other parts of the boiler to be so constructed that the boiler will be allowed a working steam pressure of 170 pounds to the square inch.

ESCAPE.

The escapement chambers will be round and fitted with rotary valves. They must be so bored and turned as to be not only steam tight, but so adjusted that they can be changed without danger while the engines are working; the chambers to be connected with T or Y pipe, provided with a stuffing box for the passage of the water supply pipe from the doctor into the main exhaust pipe to the chimneys; these valve chambers will connect, one on each heater head; the main exhaust pipe will run from this connection to after end of boilers, where a similar connection, either T or Y, will allow the exit of the water supply pipe, and from two branches of exhaust pipe, one passing into each chimney; the chambers on side of boat for outboard exhaust and to wheel, will be of the ordinary slide and butterfly valve style.

NIGGER ENGINE.

One auxiliary engine, with cylinder no less than 5½ inches in diameter and 14 inch stroke, connected with capstan by upright shaft, 2½ inches diameter; journal and pinion bearings turned 2⅜ inches; fore-and-aft shaft 2¾ inches diameter, turned to 2⅝ inches for journals and pinions; good, strong cast-iron gear wheels and pillow blocks; one cast-iron, double body capstan, for double and single purchase, with wrought-iron spindle, 4½ inches diameter, and wrought-iron counter shaft, as long as possible; pawl bed and counter shaft bush to be cast together, and the main bush to be made separate, of square form, and fitted into a similar opening in pawl bed, with surplus metal allowed on the casting for adjusting centers; the steps for lower end of spindle are also to be cast together and bolted down to breast hook; the deck at capstan to be tied to breast hook to prevent its being raised by the capstan in case of under strain on the gearing; a ratchet capstan is to be fitted and placed for a stern capstan, with sheaves in engine room for the same.

PIPES.

All pipes to be made of wrought-iron, except when otherwise specified. Exhaust pipes to be made of hard rolled copper, No. 16 wire gauge; from engines to heaters to be 6 inches in diameter; from heaters to outboard chambers, 6 inches; uprights, 8 inches, and wheel pipes, 5 inches in diameter; main exhaust pipe to chimneys, 7 inches in diameter, and provided with supports for supply pipe; and branches from after end of boilers to chimneys, to be made of hard, rolled copper, 35 pound sheet, 4½ inches in diameter, with 12 inch billy pipes of copper, 40 pound sheet; the chamber and billy pipes to be provided with

gas pipe drains, and to be fitted with receivers riveted in to receive the drain pipe; this must be large enough to receive all the water that may accumulate in the pipes; all joints to be of gum.

The main steam pipe shall be of wrought-iron of standard thickness, and have an inside diameter of not less than 3½ inches, and to be connected at terminal and intermediate joints with good and substantial malleable-iron flanges, which flanges are to be truly faced and bored to fit the pipe, slightly tapering toward the face of the flange; the ends to the pipes are to be expanded into the flanges and substantially beaded into a recess. This pipe is to be provided with a globe valve near the boiler connection.

The boiler and steam pipes will be covered with the best non-conducting material.

The main steam pipe, near the throttle, will be bent to a radius of not less than 2 feet on inside of bend.

The branch steam pipes will be of wrought-iron of standard thickness, and have an inside diameter of not less than 3½ inches, and be bent at throttle and cylinder ends to a radius of not less than 2 feet, and be expanded and beaded into malleable-iron flanges, in a manner similar to that of the main steam pipe.

The supply pipe will be 2 inches in diameter, with heavy couplings, and will pass through the exhaust pipe to after end of boilers; the connection from doctor to stuffing-box chamber aft, and from forward stuffing-box chamber to check valve on boilers, will be made of 55 pound copper, the ends flanged and provided with sleeves brazed on ends of pipe, sleeves to be of 80 pound copper, turned with the flange of the pipe, and both brazed together; the connection from forward stuffing-box chamber to check valve on boiler to contain a goose neck of sufficient radius to allow for ample expansion and contraction; the supply pipe to discharge through a Snowdon heater, or its equivalent, in the boiler.

The steam pipes for doctor, nigger engine, and whistle, to be made of wrought-iron, and to have extra heavy globe valves; the couplings of these pipes to be of brass; globe valves on all steam pipes to be finished; the steam pipe to nigger engine and to whistle will be provided with bends of sufficient radius to allow for easy and sufficient expansion and contraction.

The overflow pipe from heater will be provided with branch pipe and globe valves, so arranged that the waste water can be discharged in shoes or overboard.

The cold water pumps to be connected to a suction pipe extending from one shoe to the other, and supplied with globe valves, in such a manner that both pumps can take water from either or both shoes at the same time; such pipe to have a diameter of 2½ inches.

There will be attached to the boilers suitable pipes and valves for conveying steam into the hold and different compartments thereof, to extinguish fire; which pipes shall have an inside diameter of not less than 1½ inches. All branch pipes leading into the several compartments of the hold of the vessel will be supplied with valves, the handles marked so as to indicate the compartment or parts of the vessel to which they lead. These valves, or their handles, shall be placed in the most accessible part of the main deck of the vessel.

There will be attached to the steam fire pump two pipes having a diameter of not less than 1½ inches, one on each side of the vessel, to convey water to the upper decks, and such pipes shall have stop cocks for the attachment of hose not less than 1½ inches inside diameter, both between decks and on the upper deck; the steam fire pump to be supplied with 2 inch pipe leading to the hold of the vessel, with stop cocks or shut-off valves attached, and so arranged that the pump may be used for pumping and discharging water overboard from the hold; and each compartment bulkhead will be fitted with valves, so as to admit water from one compartment to the other, and the valves so arranged as to be worked from the main deck.

A bleeder pipe of suitable dimensions will be connected with heaters, for the purpose of heating feed water and controlling steam pressure in boilers.

AUXILIARY FEED PUMP.

In addition to the doctor, there will be an injector or steam pump of ample capacity for supplying the boilers with feed water, and such injector or steam pump shall have discharge pipe attached to the main supply pipe, and a flanged bronze or brass-seated stop cock or valve will be attached to the boiler between the boiler and the check valve.

The exhaust pipe in each chimney will have a cast-iron nozzle attached to each, and such nozzles will be contracted to a diameter of 3 inches.

BOILERS.

There will be three horizontal two-flue boilers, each 40 inches in diameter and 24 feet long, with each flue 14 inches in diameter; the shells to be made of homogeneous steel, having a tensile strength of not less than 65,000 pounds per square inch; each sheet will be tested in an approved standard testing machine, to determine its tensile strength and ductility; and each test piece shall show a contraction of area of not less than fifty per cent. at point of fracture; the circular seams in each boiler will be single riveted, and longitudinal seams will be double riveted, and be placed at least 2 inches above

the fire line, which fire line will be 2 inches above the top of the flues; all the rivet holes in the boilers will be drilled, and the boilers will be so constructed in all of their parts that they will be allowed a working steam pressure of 165 pounds under the United States law.

The flues will be 15 inches in diameter and made of wrought-iron, lap welded, in sections not to exceed 3 feet in length from center to center line of rivets in circular joints, and to have a thickness of material of not less than $\frac{29}{100}$ of an inch.

One 16 inch steam drum, with a thickness of material of not less than $\frac{25}{100}$ of an inch, and mud drum having a thickness of material of not less than $\frac{5}{16}$ of an inch. The material in each will be homogeneous steel, with a tensile strength of not less than 60,000 pounds per square inch. The legs of the steam drum will each have an area of 1 square inch for every 2 square feet of effective heating surface in each boiler. One-half of the flues and all other fire surface will be computed as effective. The longitudinal seams in steam and mud drum will be double riveted, and all rivet holes will be drilled. The flanges of the legs of the mud drum and steam drum will be double riveted both to drums and to shell of boiler. There will also be attached to each boiler one stand pipe, 14 inches in diameter, having a thickness of material $\frac{5}{16}$ of an inch. On each end of mud drum a wrought-iron nozzle will be riveted level with the bottom, and one on after side of stand pipe, low down; also nozzle riveted on stand pipe on each side of boiler for check chamber of supply pipe from deck pump; two nozzles on steam drum, one for main steam pipe and one for steam pipe for nigger engine; each end of the mud drum to have a suitable blow off or mud valve attached.

Each boiler will have a man hole in the upper part of the after head, and such man holes shall have an opening of not less than 11 x 16 inches in the clear, and be stiffened around the inner edge by a ring having an area of material equal to the amount of material cut from the head in making the man hole; and such ring will be substantially rivited to the head. Or in lieu of such stiffening ring, the man holes may be flanged to a depth of not less than $1\frac{1}{2}$ inches inwardly, measuring from the outer surface of the head, and such flange shall be planed to a smooth and even surface.

Each boiler head to have a thickness of material of not less than $\frac{9}{16}$ of an inch; each steam-drum head and mud-drum head shall have a thickness of material of not less than one-half inch. The forward head of each boiler at the lower part thereof, and each mud-drum head at the lower part thereof, and each stand pipe in the lower part of the shell thereof, and each steam-drum head, will be supplied with a hand hole of suitable size, with plate complete.

Each boiler head will be substantially stayed with good and substantial braces each $1\frac{1}{4}$ inches in diameter and attached to the heads and shell with two $\frac{7}{8}$ inch rivets of the best quality in each end of brace. Each head to have a sufficient number of such braces, and so spaced from center to center of braces as to produce a strain on each brace not to exceed 6000 pounds per square inch of section on braces.

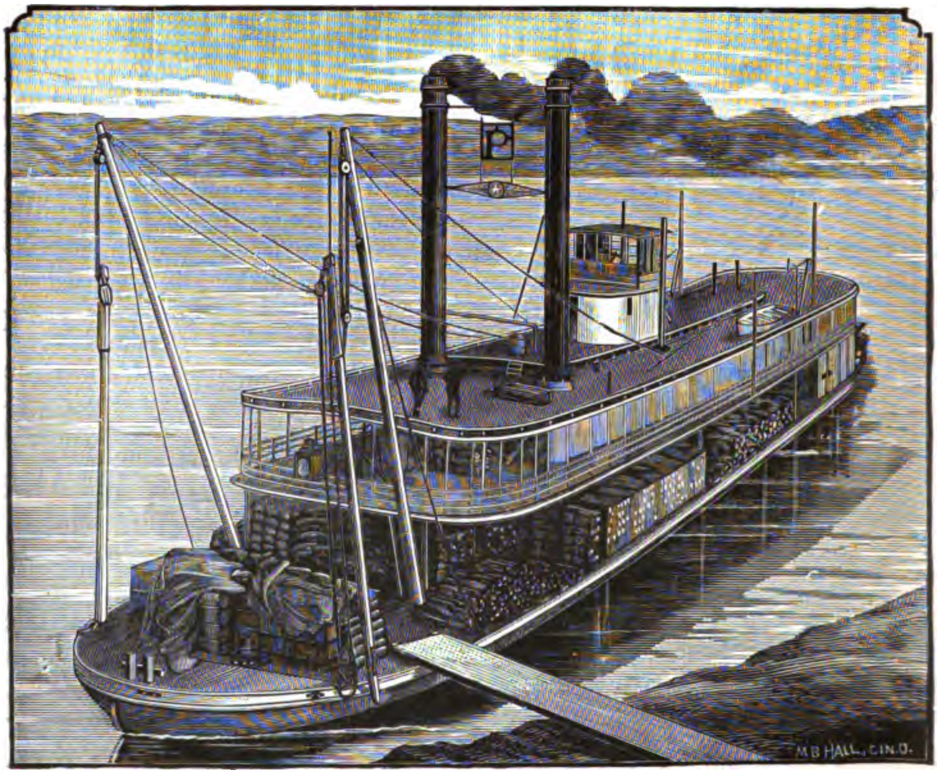


Fig. 239.
OLD-TIME MOUNTAIN BOAT ON THE MISSOURI RIVER.

Each boiler will have attached to the upper part of the shell at the center of the second sheet from the forward end, or at a distance from such forward end not to exceed 6 feet, a lever safety-valve, made in compliance with the specifications of the United States law; and each such safety-valve shall have a diameter of not less than $3\frac{5}{8}$ inches, or an area of not less than 1 square inch for every 2 square feet of grate surface of each boiler.

Each outer boiler will be supplied with three gauge cocks; and middle boiler will be supplied with two gauge cocks; and all such gauge cocks shall be of the Mississippi gauge cock pattern, or its equiv-

alent. The middle gauge cocks in the outer boilers will be not less than 4 inches above the top of the flues; and in addition thereto each boiler will be supplied with a good, substantial and reliable low-water gauge of approved pattern. There will be two steam gauges attached to the boilers, one in the engine room, and one in the fire room, and each to be so constructed as to correctly indicate the steam pressure up to and including 200 pounds per square inch.

The boilers will be placed 10 inches apart, and will be allowed a space of 10 inches between the fire bed and the shell of each outer boiler.

FURNACE AND FIRE FRONTS.

The furnace will be fitted with the most improved grate bars, with the addition of one extra set complete. The grate bars will be 4½ feet in length, and set at a distance of 16 inches from the bottom of the boilers.

There will be cast-iron fire fronts of the most improved pattern, with furnace and ash-pit doors and liners complete.

FIRE BED.

The fire bed will be formed with rounded corners at sides and after end, to avoid square corners in the draught courses, to be no less than 8 inches from boilers at bridge wall, and 12 inches at stand pipe, and to come up well on sides of outer boilers; to be supported with tee bars at intervals of 3 feet; and braces on sides 30 inches apart, provided with iron pedestals and adjustable screws to allow being raised and lowered 1½ to 2 inches. The fire bed to be of iron of No. 12 wire gauge, and lined with fire brick, in accordance with the most approved practice, and paved on bottom with common brick.

FUSIBLE PLUGS.

Each boiler will have inserted two fusible plugs of the kind and dimensions specified in the United States law. One such plug will be inserted in the shell of each boiler not less than 4 feet from the forward end and 2 inches above the level of the top of flues; and one will be inserted in the top of one of the flues at the after end of such flue.

BREECHING AND STACK.

The breeching will be supported by 1½ inch pipe braces, and be made of No. 12 iron, formed of sheets fairly and neatly riveted together, and to contain doors so arranged as to facilitate access to the flues for cleaning and repairing. The chimneys shall be two in number, and each have a diameter of 30 inches, and from breeching to hinges, for

lowering the upper part of chimneys, to be of No. 16 iron, and the upper part of chimneys to be of No. 18 iron. The stumps and chimneys to be stiffened with three bars $1\frac{1}{2} \times \frac{3}{8}$ inch, and provided with suitable hinges, stump braces, connecting bars and guy rods. The chimneys to be hinged above the hurricane deck and fitted with good and efficient lowering apparatus, complete in every respect.

BOILER CLEANERS.

Each boiler will be fitted with a boiler cleaner equal to Sims' or Sharps', with all pipe and valve connections complete and ready for use.

MATERIAL AND WORKMANSHIP.

All the material and workmanship of the engines, boilers and their appurtenances are to be of the best quality, and the whole erected and completed in a good, substantial and workmanlike manner, equal in every respect to similar work of the first class, and to comply with all of the requirements of the United States laws relating to the construction and equipment of steam vessels, and to the satisfaction of the superintending engineer.

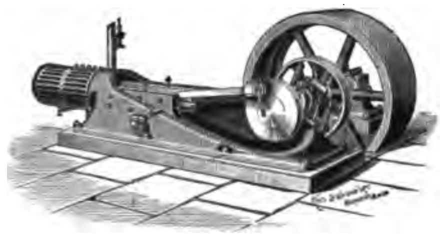


Fig. 240.
WESTERN RIVER ELECTRIC LIGHT ENGINE.

PLANS OF CONSTRUCTION FOR NON-CONDENSING ENGINE FOR A WESTERN RIVER SIDE-WHEEL STEAMER.

This type of engine is in common use on Western rivers. It is of the ordinary puppet valve type, with the exception that it is provided with a tripping gear for adjusting the point of cut off, and in addition thereto is supplied with dash pots A, A, as shown in Fig. 248. The puppet valves are worked by the ordinary "cam and lever" movement, with separate cams for steam and exhaust valves.

FEED-WATER HEATER.

Figs. 252 to 257, inclusive, show a type of feed-water heaters in common use on Western river steamers.

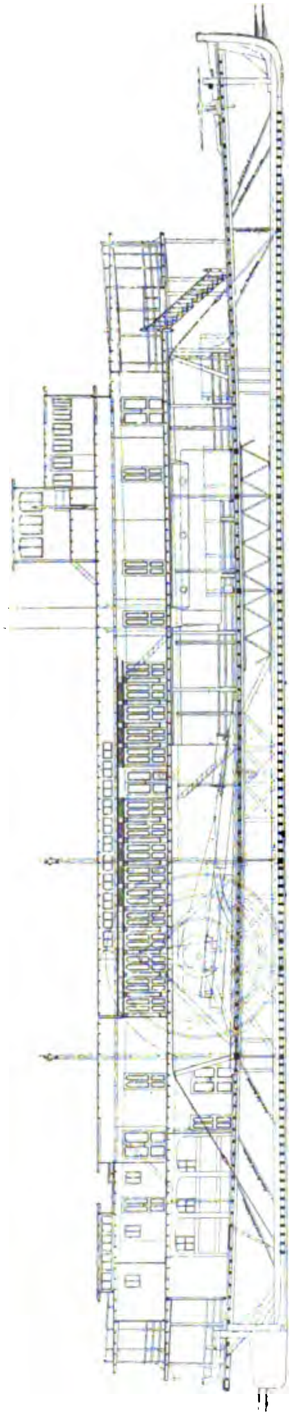


Fig. 241. Side Elevation and Section Through Center Line

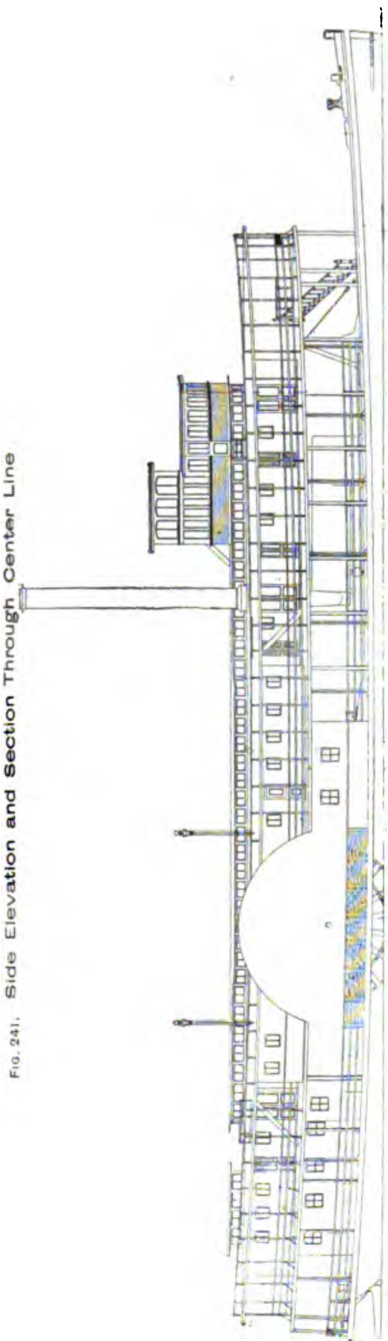


Fig. 242. Side Elevation Above Draught Line.

Two heaters are required for each steamer—one for each engine. Each heater contains 55 tubes, 2 inches in diameter and 12 feet in length, set in tube sheets J (Fig. 255), $\frac{9}{16}$ of an inch apart. The shell A is made of wrought-iron $\frac{3}{16}$ of an inch thick, and attached by means of angle-iron flanges riveted to heads B and C, as shown in Fig. 253.

The waste pipe I, shown in Figs. 252 and 253, is made of one inch iron pipe, screwed into the bottom of the heater, and arranged to discharge into the ash pit of the boilers or overboard.

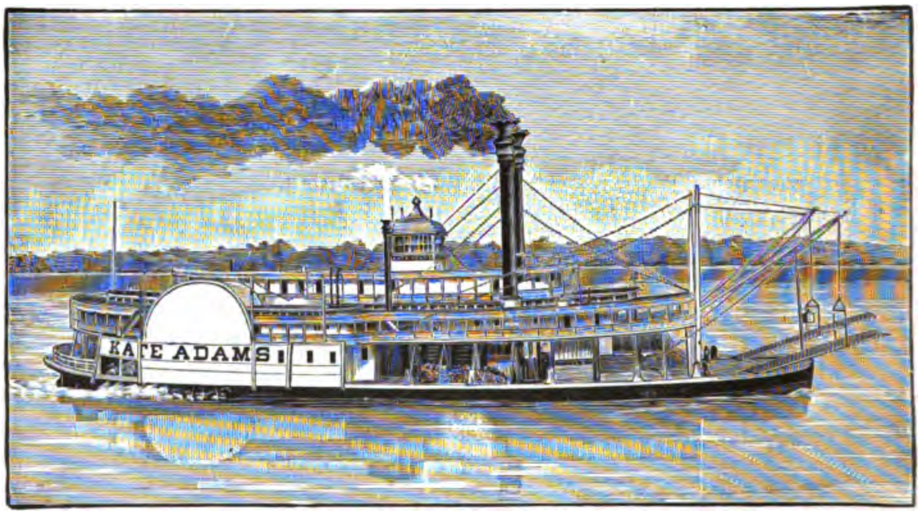


Fig. 243.
WESTERN RIVER SIDE-WHEEL STEAMER.

The exhaust steam inlet and outlet are shown at G and H (Fig. 253).

The blow off, connecting at L (Figs. 252, 253, 256 and 257), leads overboard.

The feed pipes connect at F and M, and lead from M to the boilers.

The nozzles, K, are made of $\frac{3}{8}$ inch pipe, screwed into the inner face of the head C, and extending into the tubes, D, 2 inches.

The plate E is made of $\frac{1}{4}$ inch iron, and serves to give the water a uniform circulation through the tubes D.

BOILERS.

The boilers, as shown in Figs. 259 and 260, are fire-box tubular boilers, and are employed mostly on the smaller class of Western river steamers. They are not limited as to thickness of material—externally fired tubular boilers on Western river steamers are—the law limiting the thickness of material in shells of the latter to $\frac{30}{100}$ of an inch.

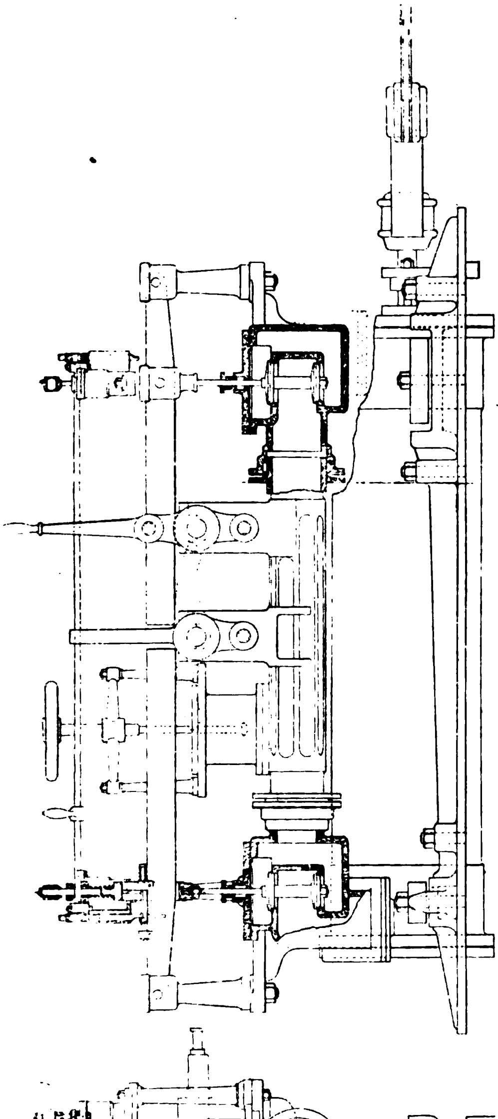


Fig 244. Side Elevation Partly in Section.

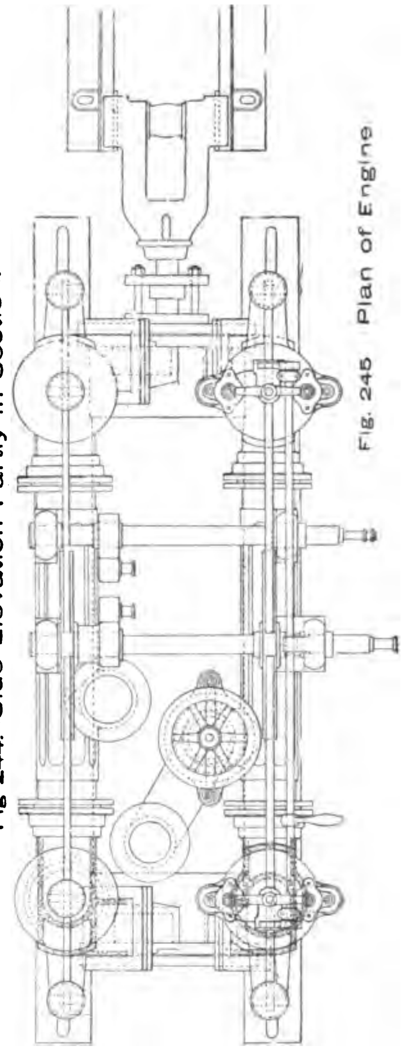


Fig. 245 Plan of Engine

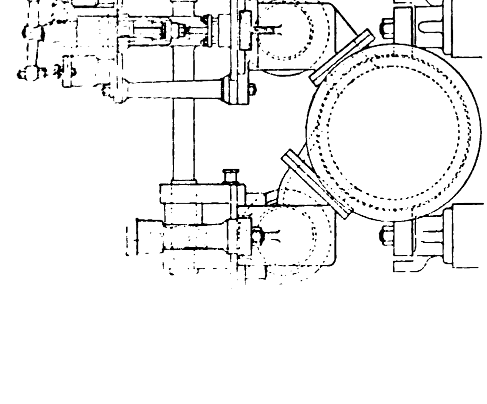


Fig. 247.

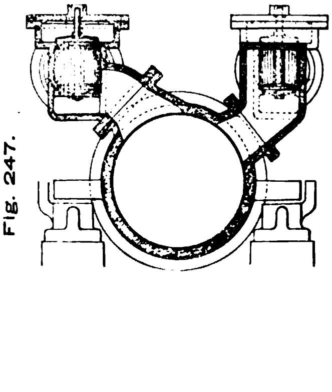


Fig. 246.

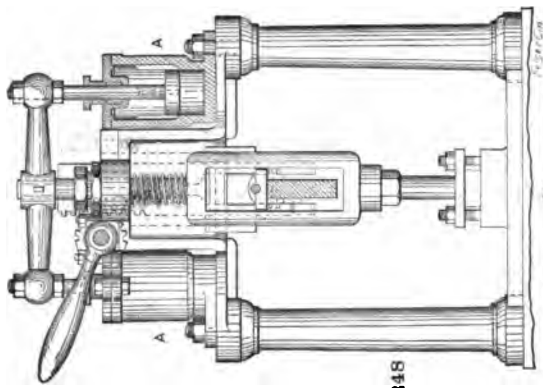


Fig. 248

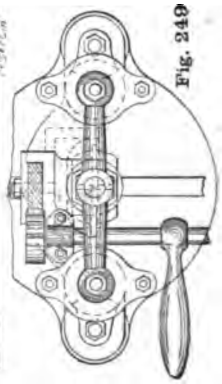


Fig. 249

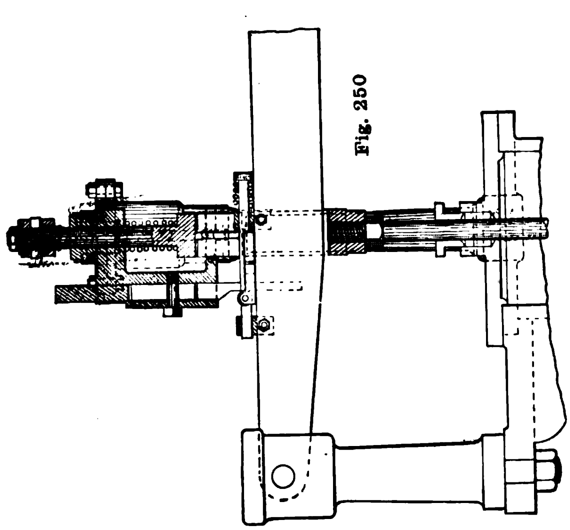


Fig. 250

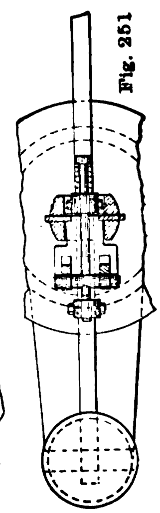
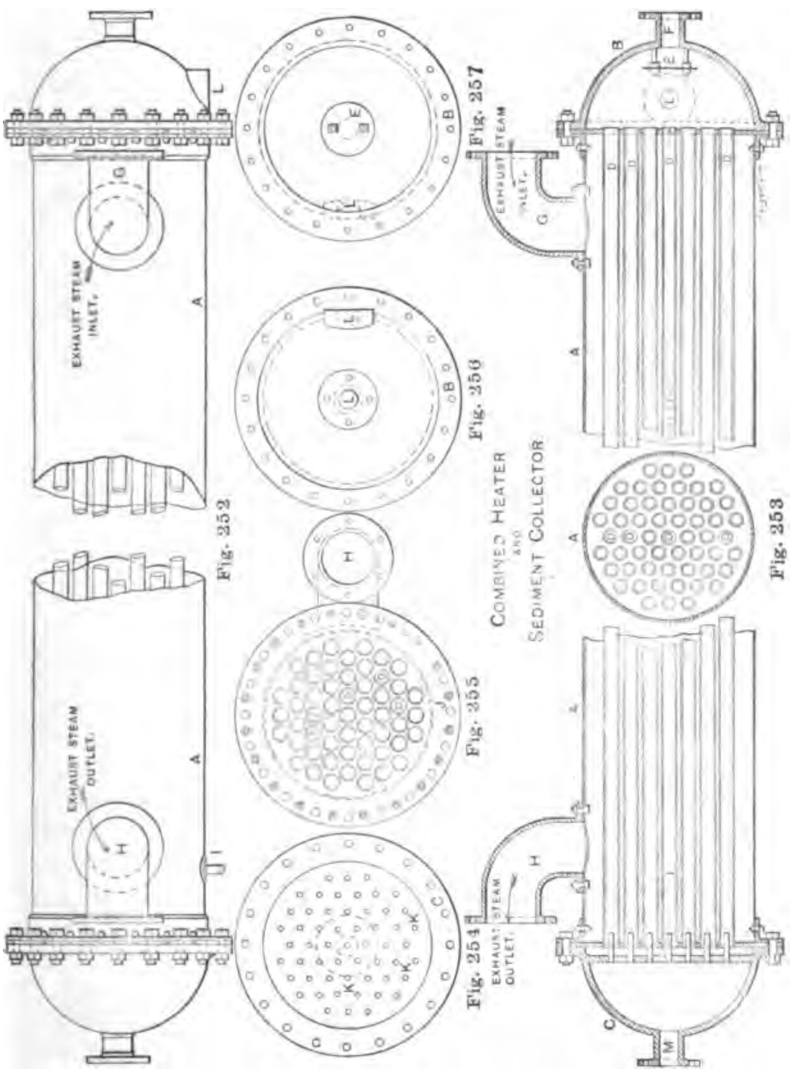


Fig. 251


EXHAUST STEAM INLET.
EXHAUST STEAM OUTLET.
Fig. 252
Fig. 253
Fig. 254
Fig. 255
Fig. 256
Fig. 257
COMBINED HEATER AND SEDIMENT COLLECTOR

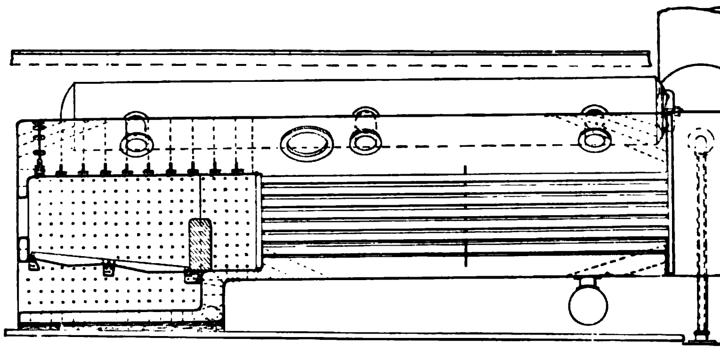


Fig. 258.

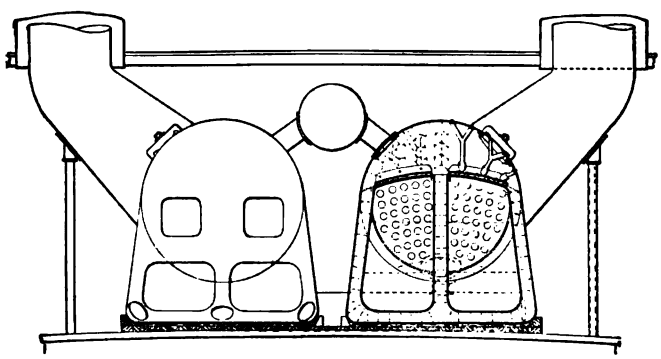


Fig. 259. Fig. 260.

THE SWEENEY MARINE ENGINES FOR RIVER STEAMERS.

The valve gear here shown was designed for the purpose of improving the operation of lever engines in the timing of their movements. It has been the practice with all lever engines on river steamers, when operated with but one full-stroke cam for both forward and backward motion, to "rider" the exhaust levers and thus improve the action of the engine, particularly on side-wheel steamers. The effect of this "rider" is merely to allow an early opening of the exhaust valves to produce an early clearing of the cylinder of the steam to make way for the return stroke of the piston. A bad feature of the "rider," however, is that for a portion of the movement both of the exhaust valves are lifted from their seats at the same time and a blow through occurs. This is not so perceptible after the engine has been put in operation and the cut-off gear engaged, because the motion of the cut-off cam is much slower than the motion of the full-stroke cam, and therefore the receiving valve, when operated by the cut-off cam, does not open quickly enough to make any perceptible blow through.

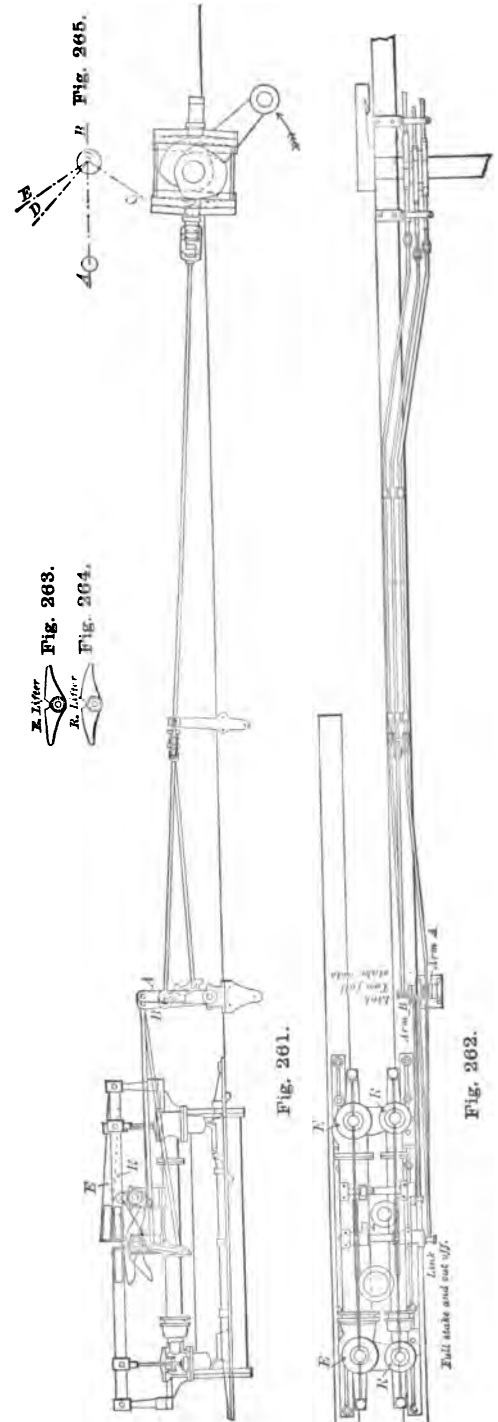


Fig. 261.

Fig. 262.

Fig. 263.

Fig. 264.

Fig. 265.

But while the "rider" of the exhaust levers has merely provided for an earlier exhaust opening, there has been no way provided for an exhaust closure; in fact, the exhaust closure is prolonged. It is, therefore, to provide for an early exhaust opening and an early exhaust closure, that the device herewith illustrated is employed.

Two full-stroke cams and one cut-off cam are employed. The full-stroke cams, originally, are set opposite to each other, and advanced to provide an early exhaust opening and at the same time an early exhaust closure. No change has been made in the exhaust valve lifter E (Fig. 263), but it will be seen that the receiving valve lifter R (Fig. 264) is materially changed in its shape. That is the points of the receiving lifter are dropped, so that while the advance of the full-stroke cam moves the exhaust lifter and through it raises the exhaust valve from its seat, the advance merely takes up the drop of the receiving lifter and puts the receiving lifter in position to begin lifting the receiving valve, while the exhaust valve has already been lifted from its seat. It is also necessary to advance the cut-off cam from its usual position with a straight lifter, so that its advance also takes up the drop of the receiving lifter. This effects a shortening in the cutting off position of the cut-off cam, as ordinarily made, and makes it possible in this valve gear to cut off steam as early as one-fourth stroke, if desired.

TO ADJUST THE VALVE GEAR.

The first part of the operation should consist in placing the cam frames in their neutral positions of motion—that is, so that each side of the frame is equidistant from the shaft. In order to do this the cams may or may not be on the shaft. Having the three cam frames so placed, adjust the length of the cam rods so that the lifters are in their neutral positions of motion—that is, so that the lifters, both receiving and exhaust, are leveled under each lever; in other words, that the points of the receiving lifter are at equal distance from the levers. Fasten the rods firmly and securely, after which place the engine on the dead center—it is immaterial which center—then put the cams on the shaft without moving the cam frames from their neutral position—that is, equally divided on both sides of the shaft.

Put one full-stroke cam opposite the other, then advance one full-stroke cam, allowing the frame to advance with it, in the direction of the forward motion, until the drop of the receiving lifter has been taken up, so that a further advance of the cam frame would lift the receiving lifter; then couple on the cam rod connecting the other full-stroke frame, and advance the other full-stroke cam in the direction of the backward movement until the drop of the receiving lifter is all taken up.

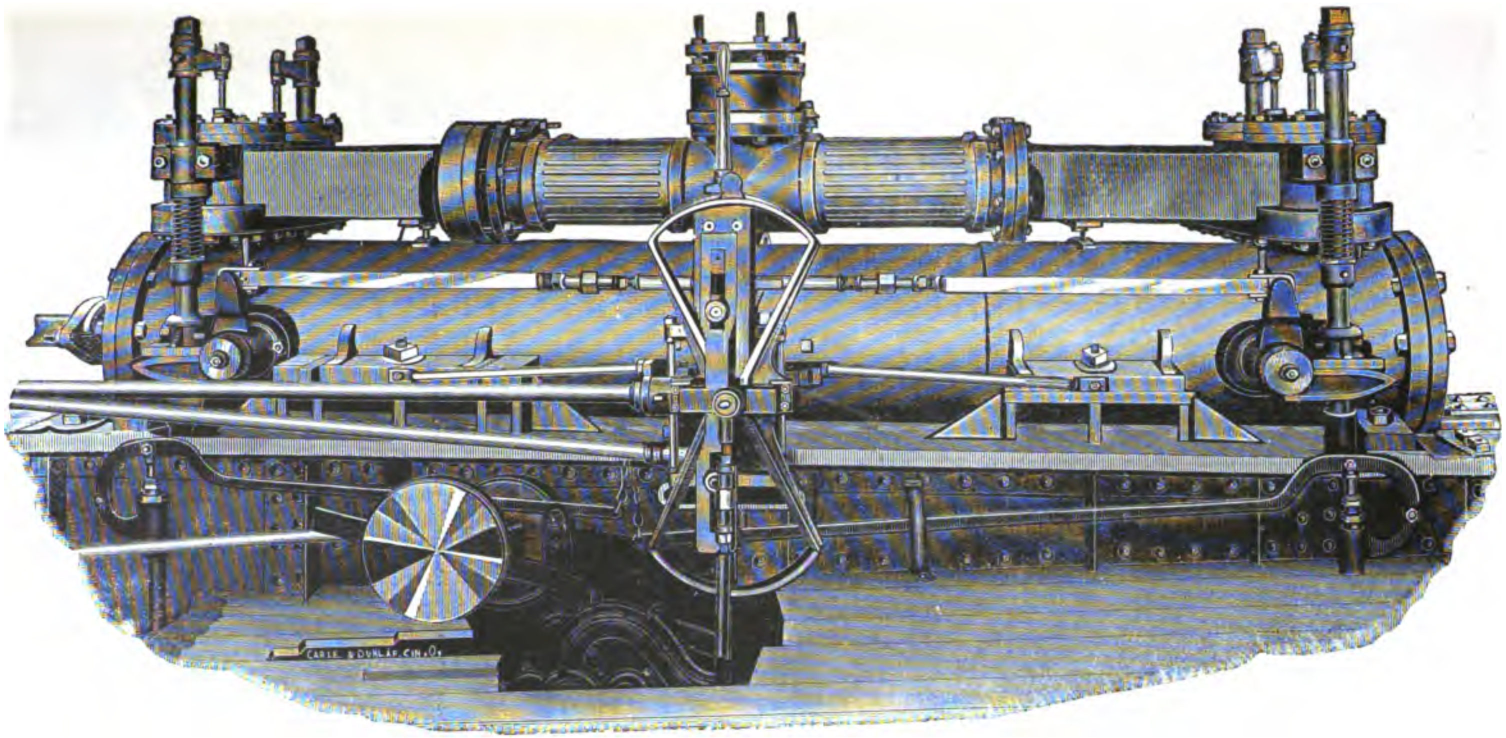


Fig. 266.

FRISBIE'S IMPROVED MARINE ENGINE.

Next put on the cut-off rod, and advance the cut-off cam, together with its frame, in the direction of the forward motion, until the drop of the lifter is all taken up; bolt the cams fast in their several positions, and the work is completed.

Care must always be taken to adjust the length of each cam rod so that it will "hook on" when the cam frame is in its neutral position, as above described.

THE FRISBIE IMPROVED MARINE ENGINE.

This engine is used largely on Western rivers, and it differs from the ordinary or typical Western steamboat engine in that the valves are operated without the use of levers. Steam is admitted to and discharged from the cylinder through the medium of balanced puppet valves, as shown in Fig. 267. The arm A operates the steam valve,

Fig. 267.

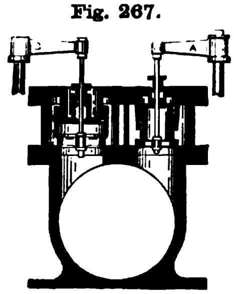


Fig. 269.

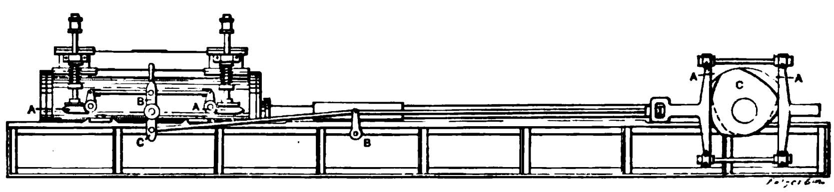


Fig. 268.

and the arm B the exhaust valve. These valves present an ingenious and at the same time effective arrangement for balancing them. Fig. 267 is a cross section through the center of the valves at one end of the cylinder. Each valve has a cylindrical projection above it, and the steam or admission valve is fitted on the inside of a cylindrical projection downward from the cover, and is made steam tight by the use of two packing rings, as shown in the illustration. The steam is admitted into the cylinder under the valve at the outer circumference after being raised from its seat. The cylinder of which the steam valve is composed has a flange projection at the lower end, and the lower outer circumference forms the seat. The flange projection at the lower end of the cylinder which forms the valve presents the only surface subjected to downward pressure from the steam coming from the boiler.

After the steam has been admitted into the cylinder and the valve is closed, the pressure under the valve is counterbalanced by the pressure of steam on the upper end of the cylinder forming the valve. The inside of the lower part of the valve cylinder is provided with arms and a hub, through which the lower end of the valve spindle passes, and made fast by means of a nut, as shown in Figs. 267 and 269. It will therefore be noticed that the steam, after the valve is raised from its seat, has free access to the inner part of the valve cylinder.

The exhaust valve differs from the steam valve in three important particulars. The first is that it is fitted to the outside of a cylindrical projection attached to the cover, as shown in Fig. 267; the second is that it has no flange projection under which its seat is formed; the third is that the pressure of steam in the cylinder, acting on the annular surface at the lower end, is more than counterbalanced by the pressure of steam on the surface formed by the shoulders on the inside of the cylindrical projection above the valve seat.

It will therefore be seen that the arrangement for balancing both the steam and exhaust valves accomplishes its purpose completely and effectively.

CAMS.

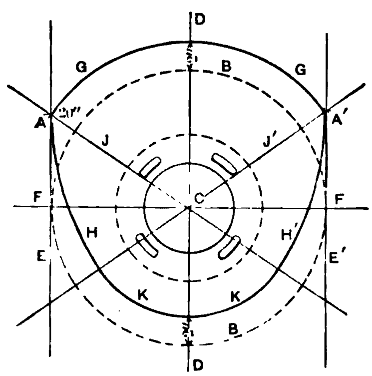


Fig. 270.
FULL-STROKE CAMS.

Cams are in general use on Western steamboat engines; which engines are slow in speed, and nothing has been discovered that will do the work quite so efficiently as the ordinary cam. It is quick and abrupt in its action as compared with the ordinary eccentric, an essentiality which is indispensable to this class of steamers, especially in making landings. The cam is adjusted to the main shaft in a manner similar to that of an eccentric, and it performs the same functions for slow-speed engines, but it is not adapted for high-speed engines.

TO LAY OFF A FULL-STROKE CAM.

To lay off a full-stroke cam, as shown in Fig. 270, make a circle 20 inches in diameter, as shown by the dotted line B B; then draw the perpendicular lines E E' at right angle with the horizontal line F F, touching the circle B B at the intersection of the horizontal line F F. Lay off the required throw of the cam from circle B B upward on line D D, which in this case is 2 inches. Place the point of the compasses in the center at C and draw the circular line G G, touching the perpendicular lines E E' at A A'. Next, set the point of compasses at A in the intersection of lines E and J, and draw the line H' from the point A' in the intersection of lines E' and J' down to the line J; then place the point of the compasses in the intersection of lines E' and J' at A', and draw the line H from the point A in the intersection of the lines E and J down to the line J'. Next, lay off 2 inches upward toward the center C, on the line D D, from the lower part of the dotted circle B B, and from that point, with the point of the compasses at the center C, draw the line K K to meet the lines H and H' at the lines J and J', and the laying off is completed.

TO SET A FULL-STROKE CAM.

Put the engine on the dead center; then set the cam at right angle with the crank. If the engine is on the forward center, set the cam with the working parts A A' up, as shown in Figs. 268 and 270. Set the rocker arms B B' straight up and down, as shown in Fig. 268; then adjust the cam rods so that connection can be made with the rocker arms without moving them. Move the spider hook, as shown in Fig. 266, from one rocker-arm pin to the other. If the rocker arm does not move, the cam rod has the proper length. If the rocker arm B (Fig. 268), has not been placed exactly horizontal or at right angle with the line of motion of the cam rod, it may be discovered by first unshipping the spider hook and then turning the engine over and placing the crank on the other dead center, and then moving the spider hook up and down on the rocker-arm pins. If the rocker arm moves, lengthen or shorten the cam rod, as may be required, one-half of the distance the rocker arm moves; then place the spider hook in proper position and again unship it and turn the engine over and place the crank on the other dead center, drop the spider hook in position and move it up and down on the rocker-arm pins, and if the rocker arm remains stationary, the adjustment is properly made; but if the rocker arm should move by changing the spider hook from one rocker-arm pin to the other, continue lengthening or shortening the cam rod, as may be required,

and turning the engine over from one dead center to the other, until the spider hook can be shipped from one pin to the other of the rocker arm without moving the rocker arm.

TO LAY OFF A CUT-OFF CAM.

In laying off a cut-off cam draw a circle F G from the center C to a diameter three times the distance from the center of the rocker arm to the rocker-arm pin; draw the circles A and B from the center C, as shown in Fig. 271. The dotted line B represents a hub on the opposite side of the cam.

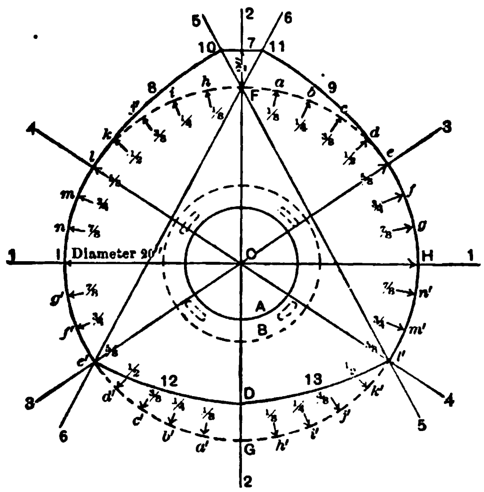


Fig. 271.

Next, draw lines 1 and 2, and divide the circle from F to H into eight equal spaces, and from F to I into eight equal spaces, and from G to H into eight equal spaces, and from G to I into eight equal spaces. We are now prepared to lay off almost any cut-off cam from one-eighth to seven-eighths, by simply drawing the lines 3 and 4 through such points on the circle between F to H and F to I to corresponding points on the opposite side of the circle, as may be desired to select; and lines 5 and 6 from F to the points selected on the opposite side of the circle; but as we are about to lay off a five-eighths cut-off cam the description will be confined to that kind of a cam.

After having divided each of the four quarters of the circle into eight equal parts, as shown in Fig. 271, draw line 3 from $\frac{5}{8}$ at *e* to $\frac{5}{8}$ at *e′* through the center C; then draw line 4 from $\frac{5}{8}$ at *l* to $\frac{5}{8}$ at *l′* through the center C; then draw line 5 from $\frac{5}{8}$ at *l′* through the intersection of line 2 with the circle at F; then draw line 6 through $\frac{5}{8}$ at *e′* through

the intersection of line 2, 5, and the circle at F; then if the valve is to move one inch, lay off 2 inches on the line 2 upward from its intersection with the circle at F; then draw line 7, from 10 to 11, from the center C, the line to be 2 inches above the circle at F; then draw line 8, from *l* to 10, with the point of the compasses at the intersection of lines 4 and 5 at *l'*; then draw line 9, from *e* to 11, with the point of the compasses at the intersection of lines 3 and 6 at *e'*; then draw line 12, from the intersection of lines 3 and 6 at *e'*, to line 2 at D, with the point of the compasses at the point 11; then draw line 13, from the intersection of lines 4 and 5 at *l'*, to line 2 at D, with the point of the compasses at the point 10. The meeting point of lines 12 and 13 must be the same distance above the circle at G as the line 7 is at F. This completes the cam as shown in Fig. 271.

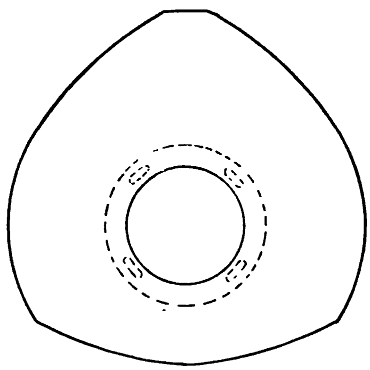


Fig. 272.

In order to avoid confusion of lines, the cam, as shown in Fig. 272, is given for the information of the student, which cam was constructed according to the foregoing directions and the illustration shown in Fig. 271.

TO SET A CUT-OFF CAM.

Put the engine on its forward dead center; then throw the cut-off cam C ahead, as shown in Fig. 268, so that it will come hard against the yoke; then adjust the rod so that the hook will drop on the pin; then tighten the collar bolts, and the cam is in its proper position. The cut-off cam may be set so as to lead a little. While this may be done with the cut-off cam, it can not be done with the full-stroke cam, for the reason that the engine is required to run both forward and backward, and the full-stroke cam is required to work the same in either case, while with the cut-off cam it is different, as it is hardly ever used when the engine is backing.

THE REES ADJUSTABLE CUT-OFF MARINE ENGINE.

This engine is used extensively, not only on Western river steamers, but also on steamers navigating South American rivers. It is, therefore, important to marine engineers that they should become familiar with its details, particularly with the adjustable cut-off, which is one of the most important features, and which is here described and illustrated for the benefit of the student of steam engineering.

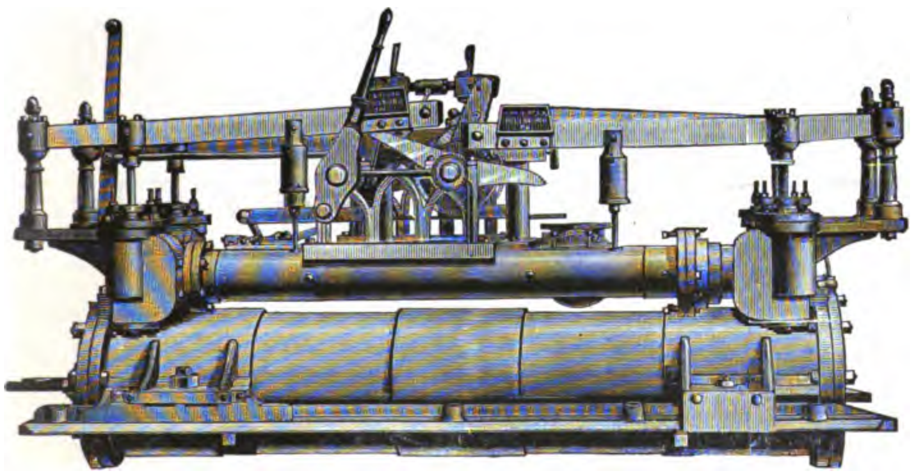


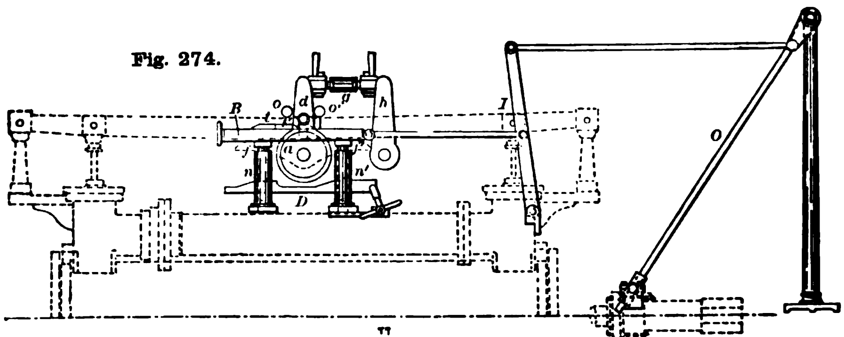
Fig. 273.

SIDE VIEW OF THE REES MARINE ENGINE.

Fig. 274 is a side elevation of cut off, and Fig. 275 is a top view or plan of the same. The full lines of the illustration show the cut off, and the dotted lines indicate the side pipes and top works, including the cross head of an ordinary steamboat engine, and shows the relative position of all the parts. The two discs *a* and *a'* are keyed fast to the forward rock shaft, to which are attached the lifters for operating the inlet valves. Between these two discs is the arm *d*, working loose on the shaft, and to which are attached, by a pin joint, the polls *o* and *o'*, which engage against the edges of the shoulders on the discs *a* and *a'*, for the purpose of lifting the inlet valves. The arm *d* receives motion through the links *g* and arm *h* on the after rock shaft, which receives its motion from the ordinary full-stroke cam. This motion being communicated through the above-mentioned parts to the discs *a* and *a'*, the inlet valves are opened and held open until the inclined blocks *t* and *t'*, on the sliding frame B, are brought alternately in contact with the polls *o* and *o'*, thus releasing them from the shoulders on the discs *a* and *a'*, and allowing the inlet valves to close; the point of

closure being regulated by adjusting the inclined blocks t and t', by a right and left-hand screw in the sliding frame B, which receives its motion through a reducing lever I, from the pendulum O, which in turn receives its motion from the cross head of the engine. By separating the blocks t and t' the steam will be cut off earlier in the stroke, and by bringing them closer together the opposite effect will be produced. When it is required to throw the cut off out of gear and work

Fig. 274.



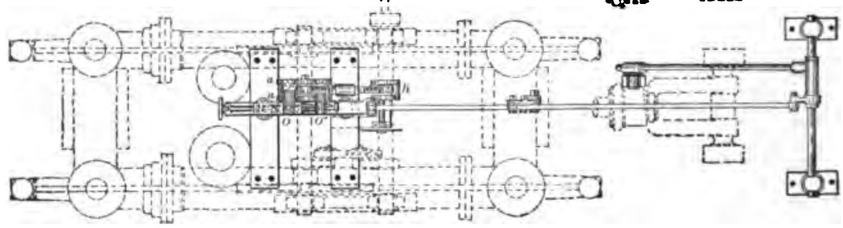


Fig. 275.

steam full stroke, the sliding frame B, carrying the inclined blocks t and t', is lowered out of the way, by moving the bar D endwise, which allows the bearings f f', of the frame B, to move down into the column n and n', thus allowing the inclined blocks to move under the polls o and o', without tripping them loose from the shoulders on the discs a and a'. The bar D is operated by hand, by means of cords attached to a lever working in the end of it and leading to the throttle valve, where they are within reach of the engineer.

It will be observed that this cut off is capable of quick adjustment, while the engine is in motion, and permits the cutting off of steam at any desired point of the stroke; besides it is simple and positive in its action, and always under easy control of the engineer. It is quick in its action and gives ample openings to and from the cylinder, with instantaneous closure of the inlet valves. It is not only capable of cutting off steam at any point of the stroke from the commencement to the end, either going ahead or backing, but it is so arranged as to

be instantly thrown out of gear, to allow the steam being worked full stroke, if desired, and can easily be adapted to existing various styles of steamboat engines.

Fig. 276 is a cross section showing the bore of the cylinder, the steam passages through the openings of the admission valves into the cylinder, and the course of the exhaust steam through the opening of the exhaust valves. The steam or admission valves being open and

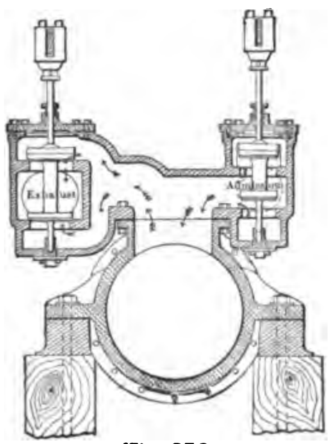


Fig. 276.

the exhaust valves being closed. These valves are the ordinary puppet valves in general use on Western river steamers. The steam pressure is downward on the upper valves and upward on the lower valves, thus counterbalancing to a certain extent the downward pressure on the upper valves. The latter being made larger than the former, so as to insure the prompt seating of the valves by means of the greater area of the upper valves, and proportionately increased pressure over that to which the lower valves are subjected.

MOVEMENT OF THE FULL-STROKE CAM.

Figs. 277 and 278 represent the engine at half the forward stroke. When an engine is at half stroke it is understood that the cross head and piston are in the center of their movement, or half way between the forward end and after end of the stroke. When the cross head and piston are in that position the crank is in the position shown in Fig. 278. This position of the crank pin is obtained by a radius equal to the length of the pitman, with the center of the cross-head pin as a center, from which is described an arc from the center of the shaft to the circle described by the center of the crank pin in revolving. By this method the exact position of the crank can be determined at any part of the stroke.

Fig. 277 is a plan, and Fig. 278 a side elevation, of the engine at half the forward stroke. The link block in link Y (Fig. 278) is down, which causes the engine to run forward, as shown by the arrows at crank Q. The cam is at its full throw aft; the cam yokes remaining at rest from the time the working point N^1 leaves the after part of the yoke until it touches the forward part, as every point between N^1 and N^2 on the working face is at equal distance from the center of the shaft. The cam rod X X is at its full throw aft, the link Y being down, making the cam rod in one straight line, gives the same motion to the cam rod between the link Y and starting arm M, causing the starting-arm pin to be at its full throw aft, making the lifter arm D (Fig. 278)—which receives its motion from arm H (Fig. 277) through the horizontal link G (Fig. 278)—at its full throw forward; and being directly opposite the starting arm M, it raises the lifter F^1 (Fig. 278), which lifts the after receiving valve lever P^1, which raises the receiving valve at that end of the cylinder and admits the steam to the after end of the piston.

The diagonal link arm V^1 (Fig. 277) is at its full throw down, the diagonal arm V on the loose lifter being down, raising the forward exhaust lever P^3, causing the exhaust valve at that end to open and discharge steam from that end of the cylinder, while the forward receiving and the after exhaust valves remain closed.

If the engine should be reversed while in this position, the link block Y (Fig. 277) would move upward and assume a position just as far forward of the center of the rocker as it is aft of the center now, making the starting-arm pin on arm M at its full throw forward, instead of aft as now shown, and just the reverse position of the valve levers would follow. The after receiving valve lever P^1 would be down; the forward receiving valve lever P^2 would be up; the forward exhaust valve lever P^3 would be down, and the after exhaust valve lever P^4 would be up; the cylinder would be receiving steam at the forward end and exhausting steam at the after end, and the engine would be moving in the direction opposite to that shown by the arrows at crank Q.

The points N^1 and N^2 are the working points or heels of the cam N. When the engine is on the forward center the point N^1 (Fig. 278) should be touching the forward part of the cam yoke, and the point N^2 the after part of the yoke. If the link block is down, as shown in Fig. 278, the point N^1 communicates forward motion to the cam rod as soon as the engine passes over the center. If the link block is up, the point N^2 gives an opposite movement, giving the engine a reverse or backward motion.

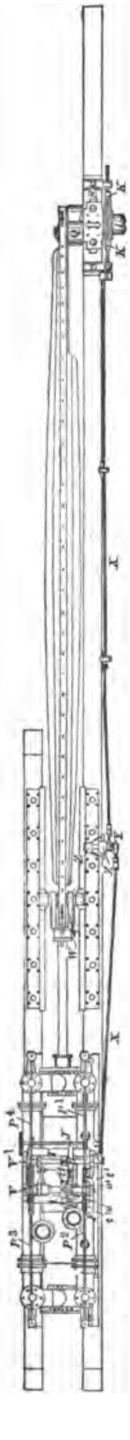


Fig. 277.

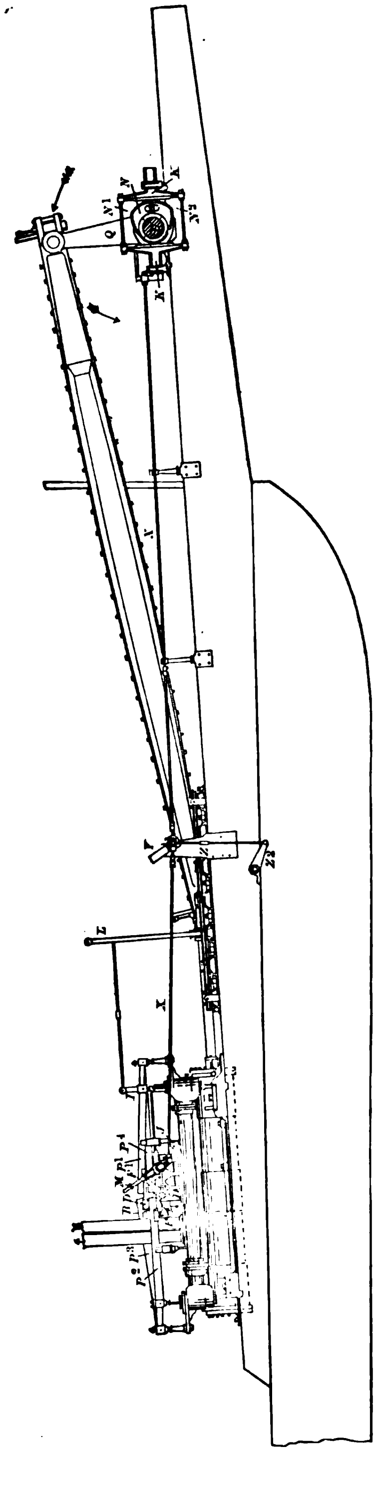


Fig. 278.

TO SET THE FULL-STROKE CAM.

This engine has but one full-stroke cam and no cut-off cam. The mechanism already described performs, in connection with one full-stroke cam, all the functions of a cut-off cam and two full-stroke cams. It is therefore necessary to set but one cam.

To begin with, place the engine on the forward dead center in the following manner: Roll the wheel forward until the cross head comes to within a short distance of the end of the forward stroke, say 1 inch. Make a scribe mark on the cross head, continuing the mark onto the slide. While the engine is in this position, make a mark on the outer wheel circle by placing a straight edge on top of the main cylinder timber; then roll the wheel forward until the cross head completes the forward stroke and makes enough of the backward stroke to allow the mark on the cross head to come to the mark on the slide. While the engine is in that position, make another mark on the wheel circle from the same position on the cylinder timber; then make a third mark just half way between the first and second marks; then roll the wheel until the third or center mark on the wheel circle comes to the same position that the first and second marks were in when made; then the engine is on the dead center, the vibration being taken out of the pitman.

The engine being on the forward dead center, set the cam square, or at a right angle with the crank, the large part, or throw, of the cam being above the shaft, or following the crank, which is done in the following manner:

Before placing the cam yoke in the bearings, set the faces the proper distance apart, allowing $\frac{1}{16}$ of an inch clearance between cam and cam-yoke faces—that is, the yoke faces are set $\frac{1}{8}$ of an inch wider apart than the greatest diameter of the cam. The cam-yoke bolts are provided with tight collars on one end, so that they can be taken apart to get them into their bearings, which is done by removing the nuts on the collar end of the bolts and putting the yokes together without further adjustment; then set the cam, by a forward movement, in such a position that the cam yoke will be central—that is, so the forward and after side will be an equal distance from the shaft; then the cam will be square with the crank, because this movement of the cam yoke is parallel to the line at which the crank is set; the faces of the cam yoke being square with this line, makes the center line of the cam square with the crank, or in other words, at a right angle with the crank, as the center line of the cam is parallel with the faces of the cam yoke when they are at an equal distance from the center of the cam or the center of the shaft.

The next point is to get the cam rod the proper length. Have all the valve levers down and the valves on their seats. After fastening the cam rod to the cam yoke, the rod should be adjusted by means of stretchers, until the engine can be reversed without moving the valve levers. If the valve levers should move, the rod is either too long or too short. If a straight link is used, such as shown in the illustration, the levers should move slightly as the link block passes from one working point to the other, owing to the straightening of the cam rod between the link and starting arm; but where a fork is used on the starting arm for reversing, it can move from one pin to the other without causing any movement of the valve levers. This completes the adjustment of the valve gear.

SPECIFICATIONS AND PLANS FOR WESTERN RIVER STEAMBOAT BOILERS AND ATTACHMENTS.

The following specifications are drawn so as to include any size of boiler that may be required by simply filling the blank spaces with the requisite data for any number, diameter and length of boilers. Besides, these specifications are made to conform strictly to all the requirements of the United States marine inspection laws. Some of the spaces are left blank, to be filled with the requirements of the law, for the reason that some of those requirements relate to particular sizes of boilers. As, for example, the thickness of boiler heads required, the size of man holes, the size of opening into boilers for steam drum leg, the diameter of safety-valves, etc. In all such cases, the engineer or student, who desires to draw up specifications for such boilers, must consult the United States law for the information required to fill the blank spaces, bearing in mind that the law limits the thickness of material for the shells of such boilers—that is Western river boilers externally fired and the heat applied to the outer shell—to $\frac{30}{100}$ of an inch, and allows no greater strain on the shell with steam pressure than that equal to one-sixth of the tensile strength of the material for single-riveted, longitudinal seams, and twenty per cent. additional where all rivet holes in the boiler have been fairly drilled instead of punched, and all other parts of the boiler are made to correspond in strength to such additional allowance.

Therefore, as the thickness of material is limited, as well as its tensile strength, the diameter of the boiler must be made in proportion to the steam pressure required. Hence, in order to be allowed a high steam pressure, such boilers are necessarily made small in diameter.

With this information, the engineer will find little difficulty in drawing up specifications for Western river steamboat boilers, if he will be governed by the following specifications:

BOILERS.

There will be —— boilers, each —— feet in length, and —— inches in diameter; each boiler to contain —— flues, each —— inches in diameter, and to be so placed in the shell that there will be a clear space of not less than 3 inches between and around such flues. The material in the shells shall be of homogeneous steel having a tensile strength of —— pounds per square inch and a thickness of —— of an inch, and a ductility of not less than fifty per cent., or in other words, show a reduction of area at point of fracture of not less than fifty per cent. The shells of boilers to be built in ——rings; and all such rings to be —— inches from center to center of holes in circular seams; and all such rings to be made in two sheets, the bottom sheet to be of sufficient length to place the longitudinal seams above the fire line. All rivet holes for the shell to be drilled with a $\frac{11}{16}$ inch twist drill for steel rivets and $\frac{5}{8}$ inch for iron rivets, and no punched holes to be allowed; all rivet holes must come fair, and no drift pin will be used forcibly. All longitudinal seams must be double riveted, and all other parts of the boilers must be so constructed as to be allowed a working steam pressure of twenty per cent. in addition to that allowed by law for single-riveted longitudinal seams. If iron rivets are used, the pitch of rivet holes from center to center in circular seams shall be $1\frac{7}{8}$ inches, and in longitudinal seams the pitch shall be $2\frac{1}{2}$ inches; the distance from edge of sheets to center of outer row of rivet holes to be not less than $\frac{15}{16}$ of an inch; and the distance between centers of longitudinal rows of rivet holes to be $1\frac{1}{4}$ inches for zig-zag riveting, and $1\frac{1}{2}$ inches for chain riveting. If steel rivets are used, the pitch of rivet holes from center to center in circular seams shall be $1\frac{7}{8}$ inches, and in longitudinal seams $2\frac{1}{2}$ inches; the distance from edge of sheet to center of outer row of rivet holes to be not less than 1 inch, and the distance between centers of longitudinal rows of rivet holes to be $1\frac{1}{4}$ inches for zig-zag riveting, and $1\frac{5}{8}$ inches for chain riveting.

HEADS.

The heads of boilers must be —— inches thick, and be of proper diameter when flanged; flange to be $2\frac{1}{2}$ inches long, and turned to a radius of 3 inches; all flanging to be machine work; the flue holes in the front heads to be flanged out $2\frac{1}{2}$ inches and to a radius of 3 inches. Back heads must have flue holes cut out large enough to admit flues, clearing rivet heads in flues; holes to be drilled and tapped in rear heads for gauge cocks and water gauges.

MAN HOLES AND HEADS.

The man holes to be made oval shaped —— inches in diameter horizontally, and —— inches vertically, and supplied with man heads and the necessary arches, bolts, and gaskets. Each man hole for the boiler proper shall be flanged to a depth of not less than 2 inches, and supplied with man head grooved to fit the flange; the flanges of man holes to be planed or otherwise brought to a true and even surface on the face of flange.

BRACING.

Each boiler head shall be substantially stayed with —— braces of $1\frac{1}{4}$ inch round iron, and to be riveted to shell not less than 6 feet from the heads with three $\frac{3}{4}$ inch rivets. There will be as many pieces of tee iron riveted to each head as there are braces; the tee iron to be $3 \times \frac{1}{2}$ inch, and each to be 6 inches long and riveted to head with four $\frac{5}{8}$ inch rivets; braces to be made double at one end, of 1 inch square or $1\frac{1}{4}$ inch round iron, and forked so as to span and bolt to tee iron with 1 inch diameter bolt.

HAND HOLES.

Front heads each to have one 4×6 inch hand hole at bottom of the head, and provided with arch, bolt, and gasket.

FLUES.

The flues will be made in sections not exceeding 3 feet in length from center to center of rivet holes in circular joints, and to have an inside diameter of —— inches at small ends of sections, and a thickness of material of —— inch; flange section on back end to be $\frac{3}{8}$ of an inch thick on flange end; flange to be turned to a radius of 6 inches; all rivet holes in flues to be drilled.

STEAM DRUM.

There will be one steam drum —— inches in diameter and —— feet in length, and a thickness of material of —— inch, such drum to be made in —— rings, with longitudinal seams double riveted; size and kind of rivets, and spacing of same, to be same as those in the shell of the boiler; the heads to be $\frac{5}{8}$ of an inch thick, and bumped to a radius of not more than 24 inches, and flanged to a radius of 3 inches; one head to contain a man hole $10\frac{1}{2} \times 16$ inches, properly fitted with a suitable man head. The inside diameter of the legs of the steam drum and

the opening into the boiler, to which the steam drum legs are riveted, shall contain an area of 1 square inch for every 2 square feet of effective heating surface in the boiler to which it is attached; one-half of the flues and all other fire surface to be computed as effective heating surface; legs of steam drum to be 10 inches in height, made of material $\frac{3}{8}$ of an inch thick, and have double-slip flanges $\frac{5}{8}$ of an inch thick, and to be double riveted to leg, drum, and boiler; all such flanges to be machine work. There will be one main steam-pipe leg —— inches inside diameter, riveted on back of steam drum, for connecting the main steam pipe. There will also be nozzles riveted to steam drum for syphons.

STAND PIPE (REAR MUD DRUM).

There will be one stand pipe —— inches in diameter and —— feet in length, and made of C. H. No. 1 iron, having a tensile strength of 50,000 pounds per square inch, and a ductility of 25 per cent.; the material to have a thickness of $\frac{5}{16}$ of an inch; the heads to be $\frac{3}{8}$ of an inch thick, and bumped to a radius of not more than 20 inches. There will be one man hole in each end of drum $10\frac{1}{2}$ x 16 inches. There will be two nozzles riveted on bottom side of drum, running aft for mud valve flanges; flanges to be $\frac{5}{8}$ of an inch thick when faced. There will also be one nozzle for deck pump, and one for donkey pump, riveted to drum, with flanges same as those for mud valves. The legs of stand pipe to be 8 inches in diameter, 15 inches long, and have a thickness of material of $\frac{5}{16}$ inch, with double slip flanges $\frac{5}{8}$ of an inch thick, single riveted to drum and single riveted to boiler; flanges to be machine work; longitudinal seams in drum to be double riveted.

MUD DRUM.

There will be one mud drum —— inches in diameter and —— feet in length, to be like stand pipe, except nozzles for mud valves, which are to be attached to ends of drum, and drum heads to have a 4 x 6 inch hand hole in each head above the nozzle for mud valve.

ASH PAN.

The ash pan to be made of $\frac{5}{16}$ inch iron, and flanged up —— inches all around, with a 24 inch ash chute extending forward of front, and to have a 2 x 3 inch angle iron along top; all holes to be counter sunk, and all joints butted with lap strips on under side of 4 x 2 inch tee iron. All rivets are to be $\frac{5}{8}$ of an inch in diameter and driven hot; the tee iron to be cut off in front, where it turns up, to allow the front part to be riveted on.

FIRE BED.

The fire bed will be made in the most approved manner; the sides, carriers, and back to be made round; sides and back to be made throughout of No. 12 iron; bottom to be of No. 10 iron; back of ash pan step $\frac{1}{4}$ inch iron; bridge wall $\frac{3}{16}$ inch iron; side bars to be $2 \times \frac{1}{2}$ inch iron; back bars to be $2 \times \frac{5}{8}$ inch iron; flat part of back and side stands $2 \times \frac{1}{2}$ inch iron; round part to be $\frac{7}{8}$ inch iron, with screw, nut, and saucer on bottom to lower and raise it; arches to be 1 inch square iron, with $\frac{3}{4}$ inch bolts; bottom bars to be made of tee iron $2\frac{1}{2} \times 3$ inches, to have five bows on each of $1\frac{1}{4} \times \frac{1}{4}$ inch bar iron, and four center stands under each bar, made of $\frac{7}{8}$ inch round iron, with thread and nut on bottom end to set in cast-iron saucer; upper end to be bolted on bottom bar with $\frac{1}{2}$ inch bolts; stands to be forked so as to straddle tee on bottom bar; the bridge wall to have same kind of bar under center, with six stands, same as others; the bottom bar to start at the head of bridge wall, and center 24 inch centers to after end of fire bed. There will be six knee braces under step of ash pan, made of $2 \times \frac{5}{8}$ inch iron, with thread and nut on same to set in cast-iron saucer, the other end to be fastened to deck with $\frac{1}{2}$ inch bolts; the fire-front braces to be $2\frac{1}{2} \times \frac{1}{2}$ inch for flat part, $\frac{7}{8}$ inch round for round part, with foot, and $\frac{3}{4}$ inch bolt through deck; side braces to be made of $\frac{3}{4}$ inch iron, with double nuts on shoulders where they go through eye or lug, riveted on one side of boilers; side braces to center 36 inches, commencing 12 inches from forward end of boilers, to within 4 inches of after end of boilers.

BLOWER SHIELDS.

Blower shields of cast-iron to be furnished for each flue.

SPARK ARRESTERS.

There will be one pair of spark arresters, made of No. 14 iron, to run from top of fire bed out to outside of ash chute, with a bar $1\frac{1}{4} \times \frac{1}{4}$ inch iron on outside for finish.

BREECHING.

The breeching will be made of No. 8 iron, and to be made round, with high round-headed rivets; the flues to be made of heavy wrought-iron, with sheet-iron doors and door liners on inside $\frac{1}{4}$ of an inch thick, with four spaces to allow air to pass between the sheet-iron door and liners; also an apron for breeching, to be made of No. 16 iron.

STUMPS.

One pair of stumps —— inches in diameter by 8 feet in length, made of No. 8 iron, with four stiffening bars on each of 2 x 2 x 4 inch angle iron, to be offset over bands, with a rivet through stiffeners and band; stiffeners to be riveted to stumps with twelve ½ inch rivets; bands on top of breeching and stumps to be made of 2½ x ⅝ inch iron, with rivets 7 inch pitch.

CHIMNEYS.

There will be one pair of chimneys, each —— inches in diameter, and 32 feet in height, made of No. 10 hard rolled chimney iron; chimneys to be made in four pieces to each set; the rings where the hinges are attached to be $\frac{3}{16}$ of an inch thick; the rings in which the eyes are riveted in for guy rods, and rings where hoisters take hold, are to be of No. 6 iron; each chimney is to have four angle-iron stiffners, with six ½ inch rivets to each sheet; sheets to be 48 inch centers; stiffners to be riveted to all bands; there will be four wire guys on each lower part of chimneys of ⅝ inch wire, with link and thumb fastenings on end; the upper end, which fastens to the eyes, to have a hook 12 inches long, made of 1 inch round iron; guy to be spliced in wrought-iron ferrule in other end of hook; top of chimneys to have two wire guys, each of ⅝ inch wire, one straight aft and one across forward, with fastenings same as the others; braces for stumps to be made of 1¼ inch round iron, with 1 inch hook bolts; the lower part of chimneys to have one cross rod, made of 3 inch gas pipe; the upper part of chimneys to have a two-cross rod—one figured bar, and one 2½ inch gas pipe; chimneys to be fitted with goose necks and sockets for signal lights; the chimneys to be provided with Christus Patent Chimney Hoister, No. 2 size; chimneys to be hinged at top of pilot house; all bands on chimneys to be 2½ x ½ inch iron, attached with rivets 7 inch pitch.

FIRE FRONTS.

There will be one heavy fire front of the most approved pattern, of the Rees or other approved make; front to be fitted with a full set of heavy liners; front and back bearing bars; the front to center on 40 inch wing pieces, to be 17 inches from the side boilers to outside of front, with 3 inch taper on each side; columns to have a large pigeon hole in each, grate bars to be 5 feet in length and of heavy pattern, and a full set to be furnished.

SAFETY-VALVES AND OTHER VALVES.

There will be one United States lever safety-valve, or one spring-loaded valve for each boiler; the spring-loaded valve, if used, must be of the pattern approved by the United States Board of Supervising

Inspectors of Steam Vessels; if the lever valves are used, the combined areas of the diameters of all the valves shall be not less than 1 square inch for every 2 square feet of the entire grate surface; if the spring-loaded valves are used, the combined area of all the valves shall not be less than 1 square inch for every 3 square feet of the entire grate surface; the valves to be of uniform diameter; the openings leading from the boiler to each valve shall in no case be less than the diameter of the valve; there will be four mud valves—two for each mud drum; the valves to be of the best make and to contain brass valve seats; there will be one check valve for donkey pump, and one check valve for main feed pipe for boilers; the valves to be of the most approved pattern, and of the best make.

WATER SUPPLY FOR BOILERS.

Each boiler will be supplied with water fed separately to the forward end by pipe and discharged in steam space; each boiler to be supplied with a Ford or Snowden heater, or other device to prevent the feed water from being discharged into boilers at a temperature less than 180° Fahrenheit; feed-water connections, where made to boilers, must be reinforced by three-fourth inch plates.

WATER GAUGES AND GAUGE COCKS.

Each boiler is to be supplied with one reliable low-water gauge of the best and most approved make; each outside boiler is to be supplied with three and each inside boiler with two efficient gauge cocks; the middle gauge cocks are not to be less than 4 inches above the top of the flues.

FUSIBLE PLUGS.

Each boiler to have two fusible plugs, at least ½ inch in diameter at the smallest end of the internal opening, and an external diameter of not less than that of a 1 inch gas pipe, one to be inserted in the top of one of the upper flues at the back end of the boiler, and the other in the shell from the inside at the fire line, and not less than 4 feet from the forward end of the boiler.

PACKING.

All joints about the boilers and fittings are to be made of usidurium rubber packing, except those of mud valves, check valves and main steam pipe, which will be Calvin joints.

BRICK WORK.

The fire brick must be laid on edge in flame bed, and center tile to be made of the Stephens' pattern; the brick in ash pan to be split brick laid in cement.

MATERIAL TO BE TESTED.

All plates subjected to a tensile strain must be tested by United States Inspectors to ascertain the tensile strength and ductility, and to receive their approval, before being used in the boilers; all rivets must be of the very best quality.

SUPERVISION.

The erection and fitting up of the work to be open to inspection at all times by any one duly authorized by the owner of the steamer for which the boilers are to be built, and his decision shall be final. Any omissions in these specifications necessary to make the work complete in all its parts, ready for steam and water connections, must be furnished by the contractor, and at his expense.

Figs. 279, 280, 281 and 282 represent plans of a battery of three Western river steamboat boilers, built according to the foregoing specifications.

The fire-box boiler, as illustrated by the following figures, is in common use on Western and South American waters. Fig. 283 is a top view; Fig. 284 is a longitudinal sectional elevation; Fig. 285 is a sectional front elevation; Fig. 286 is a front elevation; Fig. 287 is a back elevation; Fig. 288 is a sectional back elevation; Fig. 289 is a plan of bottom of fire-box legs. This type of boiler is admirably adapted for use on Western river steamers, especially for the smaller class of light-draught vessels. It is well stayed and braced, and so constructed throughout as to thoroughly meet all the requirements of the United States marine inspection laws.

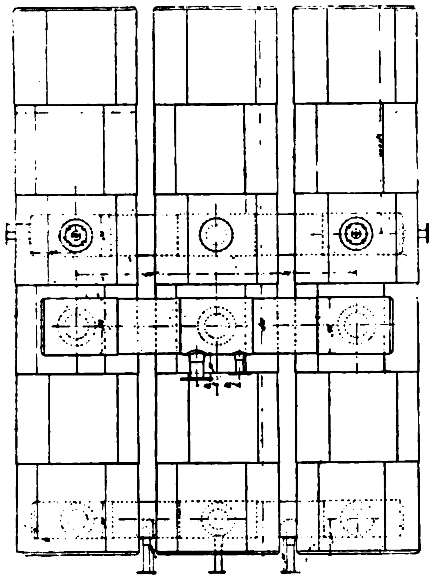
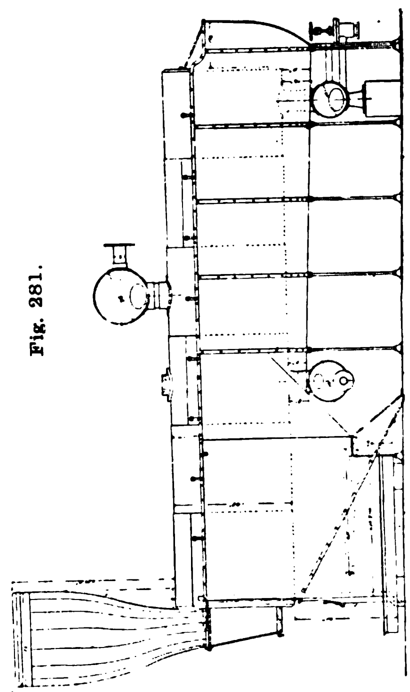


Fig. 281.



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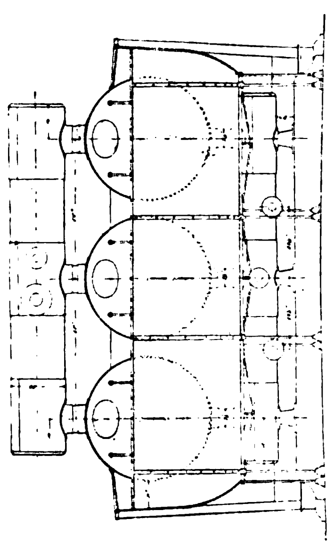

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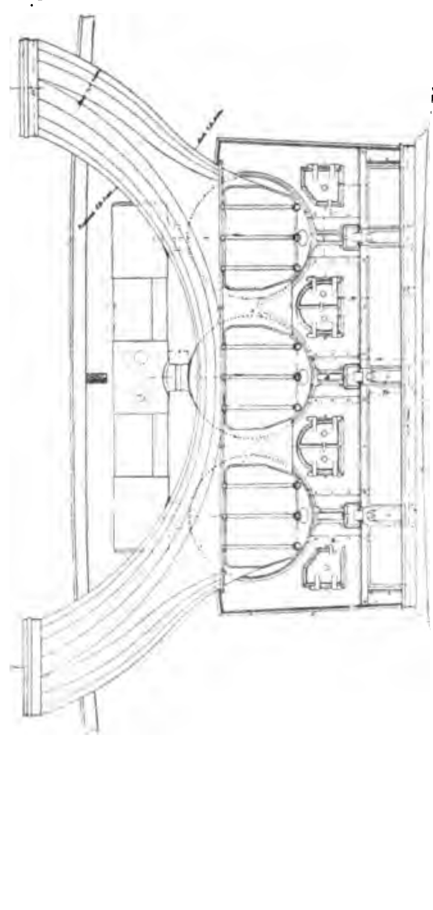

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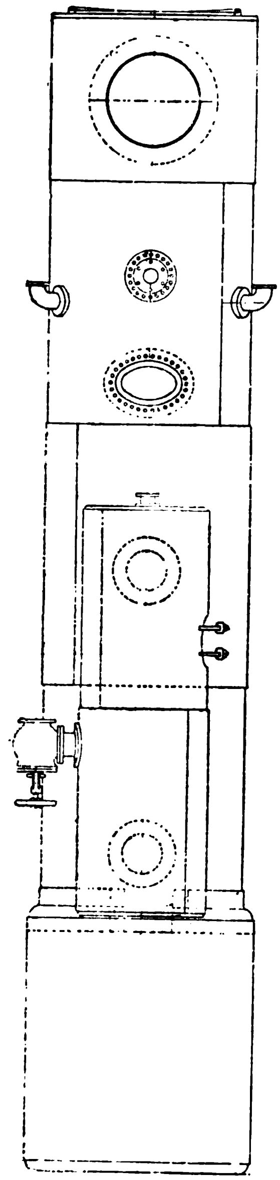

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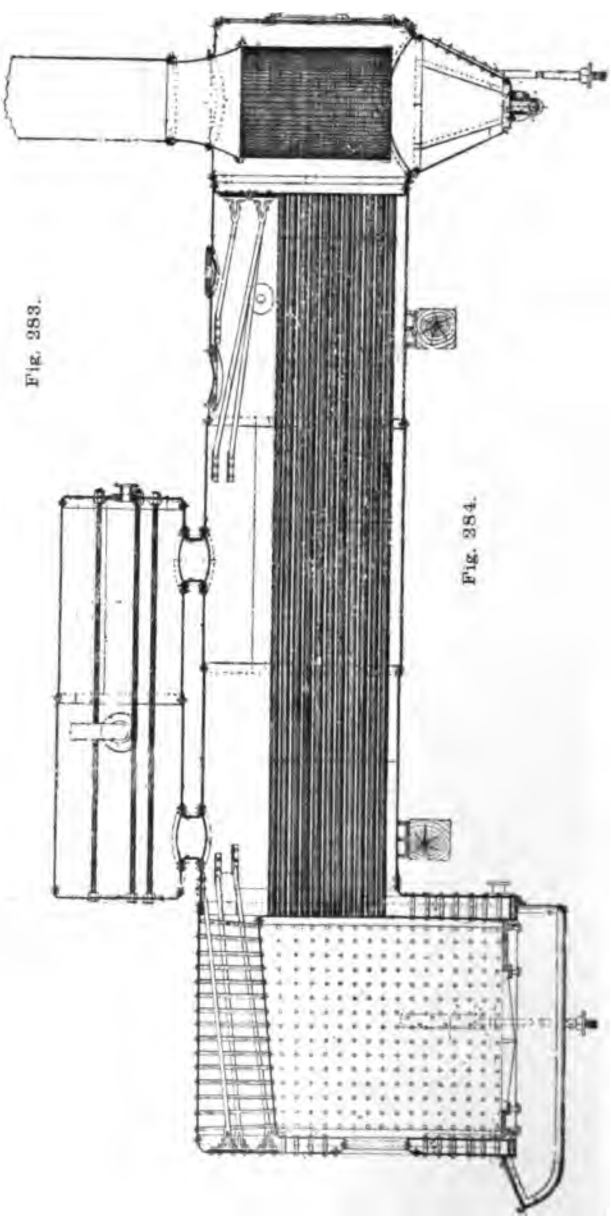

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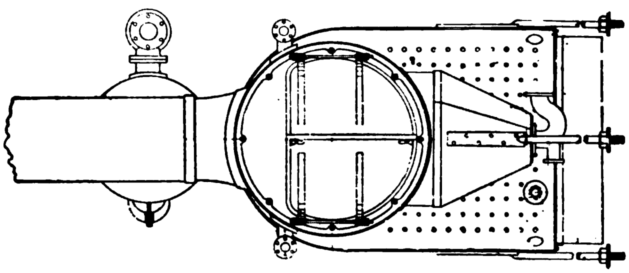

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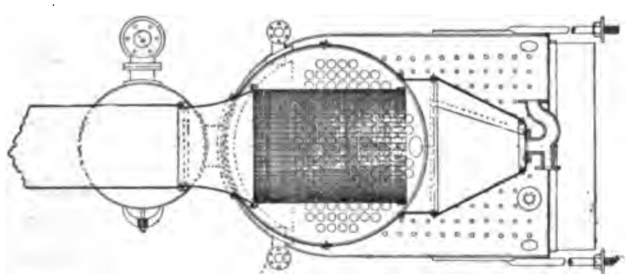

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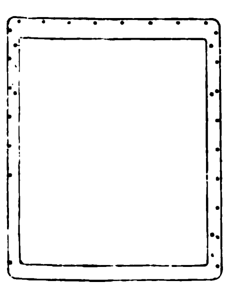

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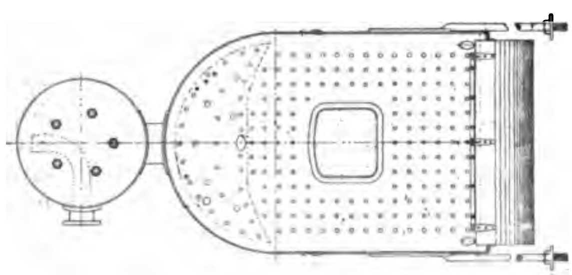


Fig. 287.

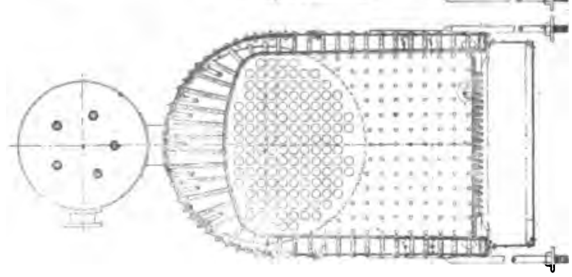


Fig. 288.

CHAPTER XIX.

THE CORLISS ENGINE.

The first of all high class steam engines, and the one which, for nearly half a century, has maintained a commanding position among the world's leading engines, was invented by that typical American engineer and Watt of the nineteenth century, the late George H. Corliss, of Providence, Rhode Island.

Since that time, and during all of these intervening years, the skill and genius of the civilized world have been engaged in the improvement of the steam engine; and yet, while the improvements of Corliss have been closely approximated, they have never been excelled in point of economy, efficiency and simplicity.

Previous to the advent of the Corliss engine, the common D slide valve, invented by Watt in the eighteenth century, was commonly employed on land engines for controlling the admission of steam to and the exhaust from the steam cylinder. The engine itself contained no means or device for cutting off steam only at some fixed point, usually about three-quarter stroke, without any regard to variations of the load to which the engine was subjected while in operation. The speed of the engine was regulated by wire-drawing steam through a valve in the supply pipe, which valve was controlled by variations in speed of the engine through the operations of a pendulum governor. In addition to this, the process of delivering steam to and exhausting it from the cylinder, through the medium of the common D slide valve, operated by the valve gear, as it then existed, was simply an aggravated form of wire-drawing steam.

The invention of the Corliss valve gear completely overcame these difficulties; and the difference in point of economy between the Corliss engine and the old type of steam engines was so marked that the Corliss rapidly superseded the other, and forged its way into every industry on the inhabitable globe; the only notable exception, perhaps, being the locomotive.

The leading features of the engine that produced such wonderful results may be enumerated as follows:

First. The employment of four valves, two steam and two exhaust valves, so placed as to reduce clearance to a minimum.

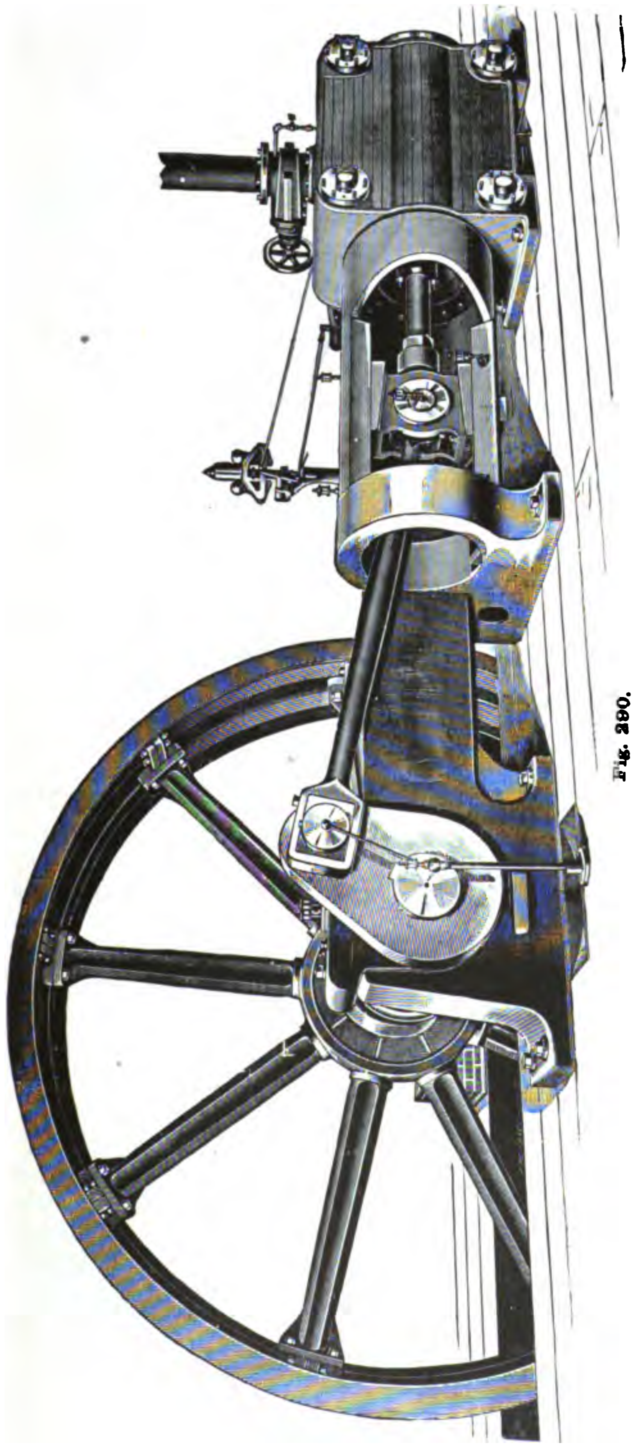


Fig. 290.

Second. The employment of a rotating valve, capable of being cheaply fitted and easily maintained, as well as being easily moved and conveniently operated by mechanism entirely outside of the steam space.

Third. The employment of a single eccentric to oscillate a wrist plate with connections direct with each of the four valves in such a manner as to cause a rapid opening and closing of the valves, and yet hold each valve open and nearly still at either end of its range, by swinging the line of connection nearly into line between the centers, and causing nearly a full port opening to be maintained during a proper interval, and thus giving a free and complete steam supply as well as a free exhaust.

Fourth. A simple and effective method for detaching the steam valves from the driving mechanism, and causing a rapid and certain closure at the proper time, so as to produce any required expansion of steam in the cylinder.

Fifth. A direct connection of the governor with the steam-valve mechanism, so as to produce the required ratio of expansion of steam in the cylinder, thus adjusting the power of the engine to correspond with the amount of work to be done, and reduce to a minimum the variations of engine speed caused by changing loads.

Sixth. Making adjustments so as to throw the least possible work on the regulating mechanism, and thus give the governor the greatest possible sensitiveness and accuracy of action.

The main points then, are the rapid and wide opening of the steam and exhaust valves; the shortness and directness of the ports and the resulting small clearances; the quickness of closure of the steam valves; the adaptation of the main valves to the functions of a cut-off valve; the connections of the governor to the cut-off gear in such a manner as to determine the point of cut-off without being itself hampered by the connections; the location of the exhaust ports at the underside of the cylinder, so as to drain the cylinder thoroughly at the end of each stroke.

Using Figs. 291, 292 and 293 for illustration, the action of the Corliss engine may be described as follows:

MECHANISM FOR OPERATING THE VALVES.

The valves are driven by the cam rod A, through the wrist plate B, vibrating on a pin C, projecting from the cylinder. Rods D D D D, having right and left screw adjustments on their ends, take motion from pins E E E E, on the wrist plate B, and transmit it to the steam valve bell crank F F, and to the exhaust levers G G, moving them with a peculiar varying motion, in such a manner as to open and close the

Fig. 292.

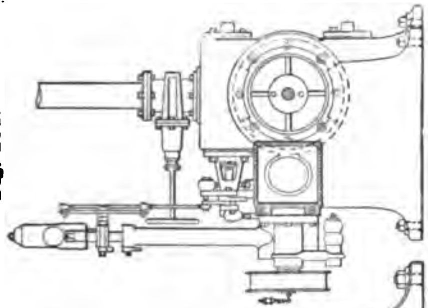


Fig. 291.

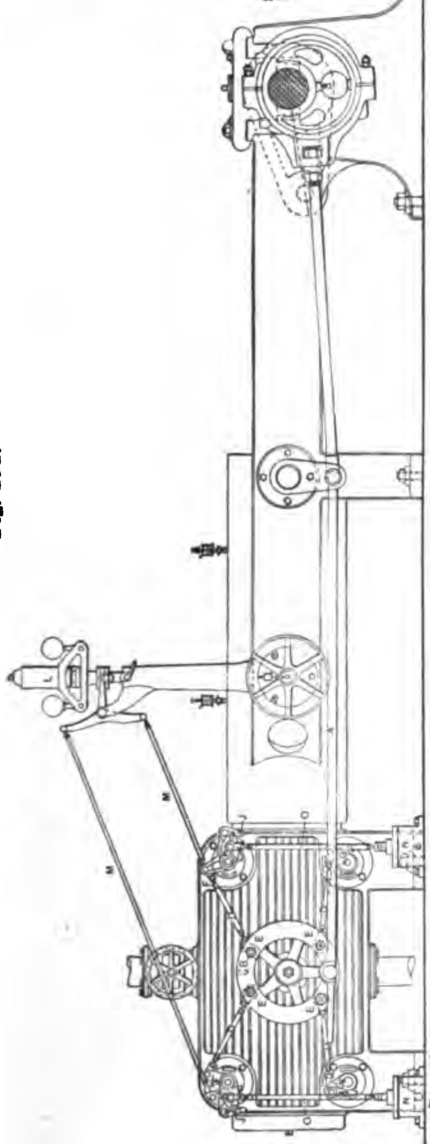
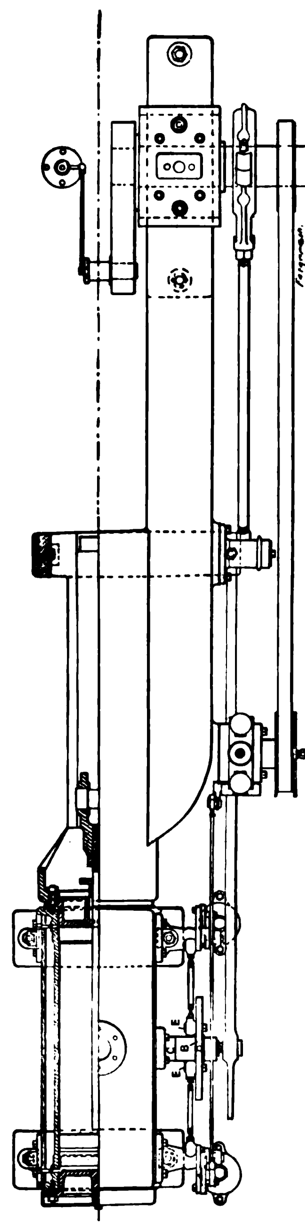


Fig. 293.



ports H H H H rapidly (Fig. 295), and to hold them open when the valves I I I I are off the ports, in such a manner as to give the least possible loss of pressure during admission and exhaust of steam.

The rods leading to the steam valves, as shown in Figs. 291 and 293, are fitted with catches, or hooks J J, which may be disengaged as the valves open, at any desired point within about half-stroke, and the time of the disengagement is determined by the rotation of the cams K K, on the valve and stems behind the bell cranks, which cams are rotated by the governor L, through the rods M M.

The slowing of the engine, in consequence of reduced steam pressure or of increased load, causes the catches J J, to hold their contact longer, and the steam to be admitted through a greater part of the stroke of the piston; and in case of increased pressure or decreased load causes the catches J J to disengage earlier during the stroke of the piston. When the catches are disengaged, the steam valves are closed by the vacuum dash pots N N, attached to the vertical rods O O, connected to the bell cranks F F.

THE DASH POT.

The dash pots are attached to the pedestals P P, under the cylinder, above the floor, in a position easily accessible. In the lower chamber of the dash pot there is a vacuum which causes the valves to close promptly. Small openings are provided for the escape of air which may leak into the chambers, and these openings are provided with spring valves which close automatically. In the upper chamber of each dash pot the compression of air is regulated by valves, which permit the quiet seating of the dash-pot plungers.

LIMITATION OF ENGINE SPEED.

Aside from the commendable feature of the Corliss engine embracing the four valves—located one at each corner, reducing clearance, permitting the steam to flow naturally through the cylinder, that is, in at the top, and with all water out at the bottom—perhaps no feature has contributed more to the success of this engine than the long strokes and moderate speed formerly invariably employed. Unfortunately, however, of late years the tendency has been to drift into too high rotative speed, and therefore too frequently at the expense of economy. Too high rotative speed means a larger percentage of clearance, an increase of friction, increased steam consumption, increase in the consumption of lubricants, and increased wear generally over long strokes and moderate rotative speeds.

The student of engineering must not conclude that high speeds are an objection, on the contrary they are desirable; it is only extreme

rotative speeds that are objected to. It must be borne in mind that whatever loss of steam results from clearance is lost each stroke, and that this loss increases in proportion as the number of revolutions of the engine are increased. Clearance therefore means waste space in an engine; and if the percentage of waste space in an engine of 2 foot stroke would be four per cent., the same amount of waste space in an engine of 4 foot stroke would be but two per cent.

FRICTION.

Again, in the matter of friction, assuming a main bearing to be 8 inches in diameter, the frictional surfaces in one revolution would move $8 \times 3.1416 = 25.1348$ inches, or an amount equal to the circumference of the bearing. Hence, whether the stroke of the engine be 2 feet or 4 feet, the main bearing friction remains the same; but the 4 foot stroke engine does twice the work in one stroke that the 2 foot stroke engine does. Therefore, a 4 foot stroke engine making 100 revolutions per minute would be doing the same amount of work that a 2 foot stroke engine would do making 200 revolutions per minute, but the latter would subject the main bearings to twice the amount of friction that the former would in doing the same amount of work. Furthermore, the wear and tear of the engine depends largely upon the number and suddenness of reversals of the direction of the motion of its reciprocating parts; and hence, the greater the number within a given time the greater the wear and tear within that time.

From what has been said on this subject the student must not be led to understand that the high rotative speed engines are objected to as such. On the contrary, many of them have been brought to the very highest stage of perfection, and they have their advantages as well as disadvantages. But what it is desired to impress upon his mind is the three cardinal points which do, or at least ought to, govern in the building of steam engines: Safety, economy and cost. First, every engine should be so constructed and so run that its speed will not menace life or property. Second, every engine should be constructed and run at a rate of speed to produce the greatest attainable amount of economy. Third, every engine should be constructed at a cost which will not be at the expense of safety or economy in running it. But in the latter lies the great fault with the construction of many steam engines. Their builders, in their eagerness to outstrip their competitors, either lose sight of or refuse to recognize the two most important points connected with the construction and running of steam engines—safety and economy. Every engine should be run within the limit of absolute safety, just as a steam boiler is run, or at least ought to be run. A steam boiler that would burst at a pressure

of 437.5 pounds, is allowed, under the laws of the United States, a working steam pressure of 125 pounds per square inch; which, if based upon the bursting pressure of the boiler, is a factor of safety of 3½; and no engineer who understands his business would be fool-hardy enough to exceed the limit. Yet, in running engines, the factor of safety is seldom taken into consideration, and in many cases they are run at as high a rate of speed as they can be made to run without considering the question of safety or economy.

In order that the student may get a correct idea of a well proportioned and constructed Corliss engine a description will be given here of one of that type of engines.

THE LANE & BODLEY COLUMBIAN CORLISS ENGINE.

This engine is produced in honor of the Columbian Exposition, and is taken as a sample of first class Corliss engines, because of the familiarity of the author with its design and construction.

This company, believing that modern progress had outgrown the forms and details of construction, which were doubtless admirable in 1849, and being desirous to exhibit an engine which would show marked improvement over the Corliss engine of forty years ago, designed and constructed the Columbian Corliss Engine. Modern conditions of steam pressure, speed, and continuity of service demanded by electric light, the manufacture of artificial ice and street railway service, required recognition. In the older forms of Corliss engines the details depended for their form and construction largely upon convenience and cheapness of production. For example, the main parts consisted of a cylinder, under each end of which was bolted a leg or pedestal. To the cylinder was bolted a girder containing the slides, the girder forming the connection between the cylinder and main shaft pedestal, or pillow block, to which the other end of the girder was bolted. This method of construction permitted comparatively simple and light separate parts, each of which could be used for either right or left-hand engines; and for the comparatively slow speeds and low steam pressures formerly employed, they answered their purpose quite well. Severer modern service, however, requires the elimination of all unnecessary joints.

The Columbian Corliss, it will be noticed by reference to Fig. 294, consists of two main parts—the cylinder and frame. The cylinder in the larger sizes being bolted directly to the foundation without the interposition of pedestals or legs, and in the smaller sizes the pedestals being cast on the cylinder. These pedestals are of box form in cross section, giving two vertical walls of metal for the direct support of the weight of the cylinder, and in addition thereto presenting to view

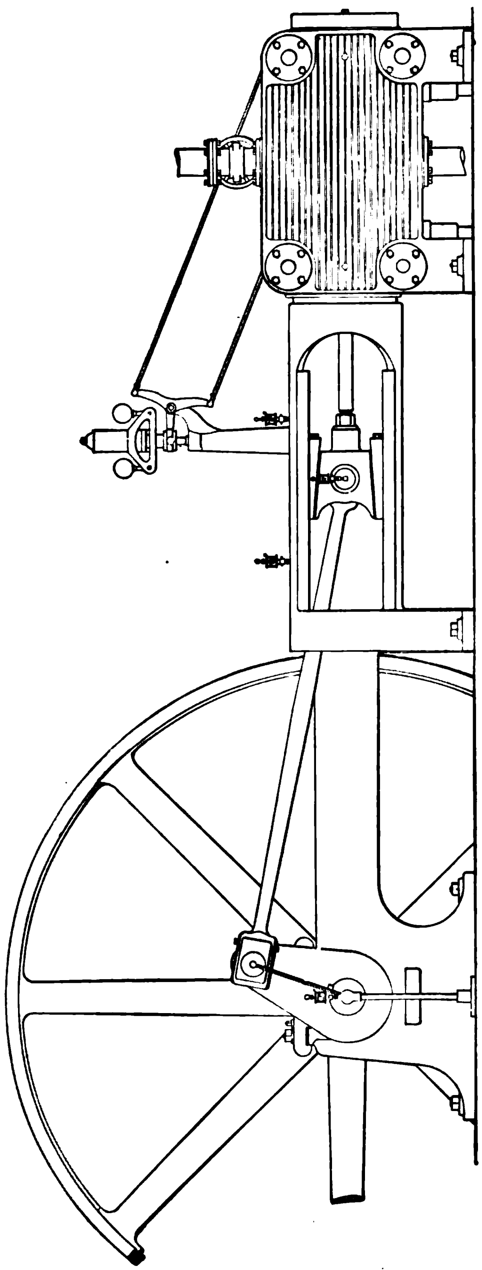


Fig. 294.

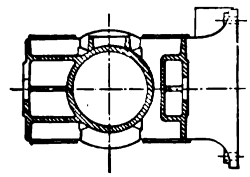


Fig. 296.

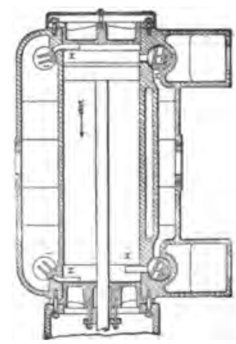


Fig. 295.

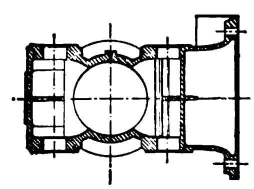


Fig. 297.

plain surfaces, easily kept clean, and affording no recesses for the lodgment of dirt. The frame has cast on its outer end the main bearing with its pedestal. This construction prevents the possibility of looseness and springing in three useless joints, besides presenting a much neater appearance. It will be noticed that the term "frame" has been substituted for that of "girder." In the older form of construction it was customary, and quite proper, owing to the comparatively light service required, to make an unsupported span from the cylinder to the main pillow block; but the severe modern service requiring a support at the end of the slide, the term "girder" in such case becomes inappropriate, and hence the substitution of the term "frame" in its stead.

THE CYLINDER.

The cylinder, as shown in Figs. 291 and 294, is fitted with circular bonnets and circular corners of large radius on top at each end; this, for many reasons, is considered an improvement over the square bonnets and square-cornered cylinder. The steam is properly guided by the interior curved surface into the ports. The absence of sharp corners on bonnets and cylinder ends renders cleaning easier and avoids the probability of bruised corners. The hole is round, and the circular bonnets and cylinder ends are in good form, besides the rounded cylinder is stronger than the square. The iron top cast on the cylinder is one of the peculiar features of this engine, and gives it a handsome finish.

LAGGING FOR CYLINDERS.

The use of wood lagging for the top of Corliss engines is not good practice. The shrinking, swelling and warping of the wood soon renders the cylinder unsightly in appearance, all of which is avoided by having the top in one piece cast with the cylinder.

STEAM AND EXHAUST VALVES.

The steam and exhaust valves, shown in Fig. 295, are placed in their proper positions in their relation to each other. Fig. 296 is a cross section through the center of the cylinder, and Fig. 297 is a cross section through steam and exhaust valves.

STEAM AND EXHAUST CHESTS.

The steam chest A, shown in Fig. 298, is much larger than such chests are usually made, as will be seen by comparison with B, as shown in Fig. 299. The object of the enlarged steam chest is to provide a reservoir for steam near the cylinder, and beyond all contracted

openings, which results in securing a better admission and steam line, shown by indicator diagrams.

The exhaust chest C, in Fig. 298, is separated from the bottom of the cylinder, and the exhaust steam with its reduced temperature, is thereby prevented from robbing the walls of the cylinder of heat.

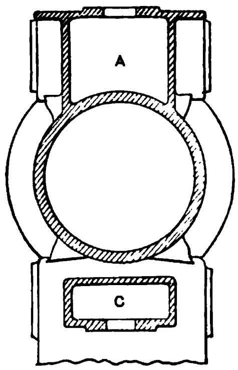


Fig. 298.

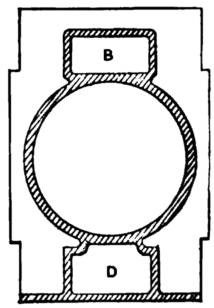


Fig. 299.

The cylinders are counter bored at each end to such an extent as to allow the piston ring to overrun the shoulder about one-half the width of the ring. The cylinders are not steam jacketed unless specially ordered by purchasers, owing to contradictory results obtained from experiments seeking to establish the value of steam jackets. Experiments recently made by men competent to obtain correct results, show that in the same engine, when worked under varying conditions of steam pressure, load, and cut off, the jacket is sometimes a source of gain and sometimes a source of loss. Assuming the correctness of these experiments, it is deemed wise to save annoyance and expense while awaiting further enlightenment on this subject.

CYLINDER HEADS.

The cylinder heads of this engine are scraped metal to metal, and a perfectly steam tight joint is produced without packing of any sort. Over the back cylinder head is fitted a plain, polished cast-iron cover, secured in position by one hexagonal head cap screw in the center. The sides of the cylinder are lagged with polished wood or iron, behind which mineral wool or other suitable non-conducting material is packed.

FRAMES.

The frame of an engine of this type is subjected to complex strains, as it combines slides for the cross head, main bearing for the crank shaft, and seats for the governor and rocker arm.

The strains between the cylinder and main bearing are tension, compression, bending horizontally and vertically, and twisting. Frames for engines of this type are usually in cross section, somewhat as shown in Figs. 300, 301 and 302.

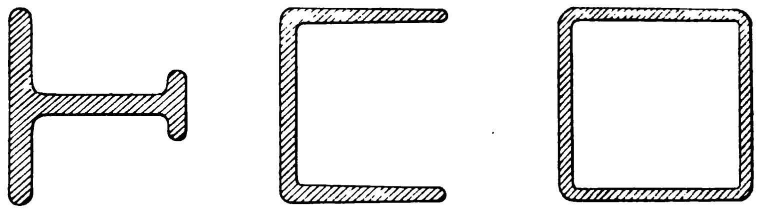


Fig. 300. **Fig. 301.** **Fig. 302.**

Fig. 300 is of proper form to resist tension, compression, and horizontal bending.

Fig. 301, or channel section, resists tension, compression, and vertical bending satisfactorily.

Fig. 302, or box section, is a form capable of properly resisting all the strains to which it is constantly subjected in service; and this has been adopted in the Columbian Corliss Engine. It must be borne in mind that if the inherent stiffness of this box section is to be retained, no side, top or bottom openings can be permitted. The form of this style of frame, in addition to its other qualities, affords no convenience for the collection of dirt or rubbish.

THE SLIDES.

The slides for the cross head are cast in one piece with the frame. In the history of the Corliss engine various forms for the slides have been adopted, among which are the forms represented by Figs. 303, 304 and 306; 305 being an exaggeration merely of 304.

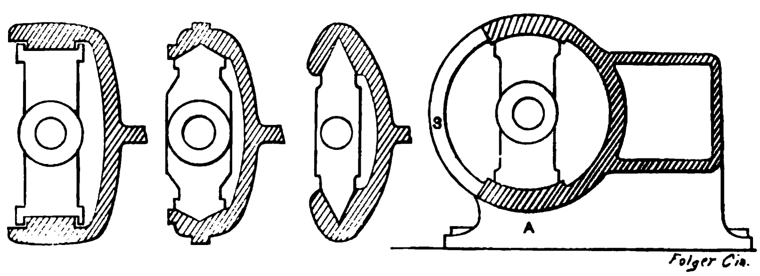


Fig. 303. **Fig. 304.** **Fig. 305.** **Fig. 306.**

The flat slide (Fig. 303), for distribution of lubricant and to obviate excessive friction, is a most excellent form; its shortcoming, however, is in the absence of any means for taking up the side wear on the inside guide plunger of the cross head.

The V shaped slides, shown in Fig. 304, is not a bad form unless the angles are permitted to run to extremes in the direction shown in Fig. 305. No side motion is permitted, but for even distribution of lubricants it is not so well adapted, as the oil tends to leave the upper edges. The friction is inclined to be excessive, and by referring to Fig. 305, which shows the V slides in exaggerated form, it readily may be imagined that a downward pressure of the cross head in the angle would produce such friction as to render it beyond the power of the steam pressure to move it.

Fig. 306, or bored guide, possesses all of the advantages of the flat or the V guides, and some peculiar to itself. It approximates the flat guide near enough to admit of perfect oil distribution, and the cross head has no tendency to wedge itself in as in the V form (Fig. 305); it affords a lateral support for the cross head when both are worn; it allows the cross head to turn slightly and adjust itself to the alignment of the crank pin should the main shaft be slightly out of level; and finally, and most important, it can be made in perfect alignment with the center of the cylinder.

The pressure of the cross head on the slide when the engine runs over on the outward stroke is downward upon the bottom slide only, for both outward and inward stroke of the cross head. This downward pressure tends to deflect the frame downward vertically, which in this engine is resisted by a support, the slide pedestal A (Fig. 306), cast in one piece with the frame, and extending entirely across the frame and downward to the foundation to which it is securely bolted by widely separated bolts. It is this pedestal which makes the term "girder" for the new design inappropriate.

The slide pedestal, as well as the main bearing pedestal, is of box section, conforming in style with that of the cylinder, each presenting two vertical walls of metal for the direct support of the loads they are designed to sustain. When the engine runs under on the outward stroke the pressure of the cross head on the slide is upward against the top slide for both the outward and the inward stroke of the piston. This tends to deflect the frame vertically upward, and to spring the top slide in that direction. In some engines there is no special provision against this. But in the Columbian Corliss Engine, at 3 (Fig. 306), at the slide end a massive brace is cast, which ties the overhanging top slide down to the bottom slide, thence by means of the slide pedestal A (Fig. 306) the strain is transmitted to the foundation, and entirely preventing the springing of frame and slide. At each end of the slide is cast a pocket for the reception of such oil as may be forced off of the slide by the motion of the cross head. The slide length is such that the cross head overruns on each end, which prevents the wearing of shoulders on the slides.

The main pedestal bearing on the foundation is narrow and very long, extent of bearing upon the foundation being secured by increase of length rather than width. The holding down bolts are two in number and in the line of the length of the engine. By this it is sought to avoid the possibility of springing the frame, as may be done with four widely separated foundation bolts in a wide base, resulting in throwing the main shaft bearing out of level. The main bearing is full babbitt lined to the extreme ends, there being neither in this nor any other babbitted bearing on the engine the iron lips at each end, sometimes employed, and which by presenting a different metal interfere with the uniformity of wear. The babbitt is thoroughly planed after being cast, and then truly bored and scraped to a bearing. On each side of the main shaft is a babbitt-lined quarter box, each adjusted by wedges extending the full length of the main bearing, and having studs extending upwards through the cap, on which are nuts, by means of which they may be raised. Since the quarter boxes receive nearly the full pressure of the steam acting on the piston, it is desirable to back them up their full length with wedges rather than to receive this pressure on the points of set screws, sometimes used for this purpose. The bottom bearing for the shaft is a part of the frame itself, rigidity in so important a bearing being imperative, and those constructed of four quarter boxes, adjustable in all directions, are the cause of much care and trouble, owing to the absence of this rigidity. The cap is provided with large oil pockets, and is so constructed as to admit of easy inspection and of feeding the shaft by hand while the engine is in operation.

PISTON PACKING.

Fig. 308.

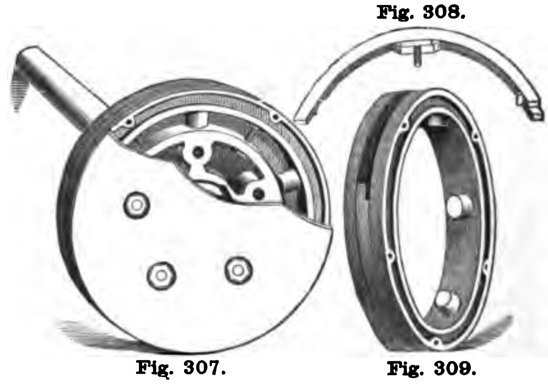


Fig. 307. Fig. 309.

The packing employed in the piston is of the Babbitt & Harris patent, which has been used for seventeen years in connection with these engines. This packing is composed of a narrow ring and is self-adjusting. Its construction and arrangement are shown in Figs. 307, 308 and 309.

TO ADJUST THE CROSS HEAD.

The requirements of a good cross head are, besides that of being of sufficient strength, that it should possess ample bearing surface upon the slides; that the distribution of the thrust from the connecting rod should be equally distributed over the surfaces; that the adjustment for wear should be such that it can be taken up evenly over the entire surface; that the slippers can be easily removed without unshipping the cross head or connecting rod; that the connecting rod can be readily detached, and that the lubrication of pin and surfaces shall be even and continuous.

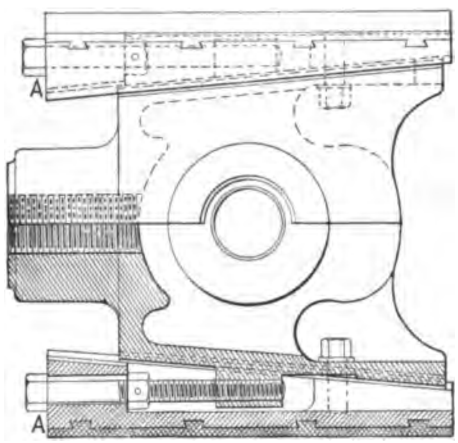


Fig. 310.

Fig. 310 shows a cross head designed to fulfill the foregoing enumerated requirements. The first is satisfied by the large surface of the slippers A A, lined with the best Babbitt metal, hammered in place before being planed. The equal distribution of pressure over this surface is obtained by having the wrist pin centrally located, longitudinally and vertically, in the cross head, and by having the cross head bear for its full length and breadth in any condition of wear upon the shoes, which are wedges and may be forced evenly forward by means of a single screw in each, located at the end of and in the middle of each shoe, the shoe being cramped in position when properly adjusted by means of cap screws. These long, tapered shoes, adjusted longitudinally, evenly taking up the wear, are superior to two or more wedges or set screws, each separately adjusted, by which one part of the cross-head shoe may sustain the whole load. By moving the clamp screws either top or bottom shoe may be readily removed endwise without disturbing any of the other parts. A tapped hole in the side of the cross head is provided for the insertion of a stud for operating the parallel motion of an indicator.

CONNECTING RODS.

The connecting rod is of the solid end type; the adjustment is effected by a block with one taper face; its position is changed by slackening one and tightening the other screw, as shown in Figs. 311 and 312. This method securely locks the wedge.

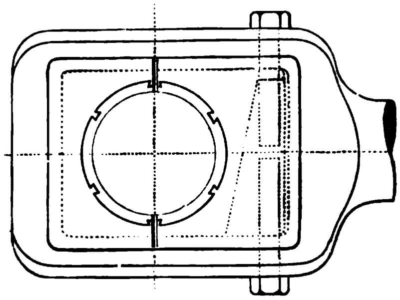


Fig. 311.

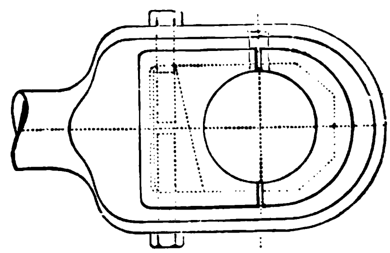
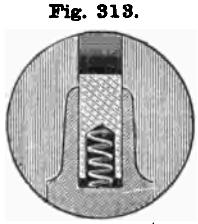


Fig. 312.

The boxes have a nearly full support by the wedge. The crank pin boxes are of bronze, lined with the best of Babbitt metal. The cross head end is shown in Fig. 312, and the crank end in Fig. 311. The cross head end has solid phosphor-bronze boxes.

VALVES AND VALVE RODS.

Fig. 313.



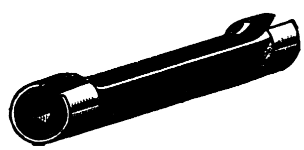


Fig. 314.

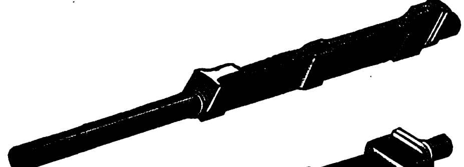


Fig. 315.

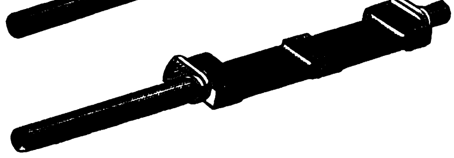


Fig. 316.

The valves are of the original Corliss type, with cylindrical ends of unusual length. The stems are of phosphor-bronze or steel, extending clear through the length of the valve, with a bearing in the back

bonnet. Spiral springs are set into the blades of the valve stems to allow some compensation. The results establish the durability of such construction, as well as its superiority over the ordinary tee head valve stem, notwithstanding the increased cost of the former over the latter. The construction of these valves are shown in Figs. 313, 314, 315 and 316.

THE GOVERNOR.

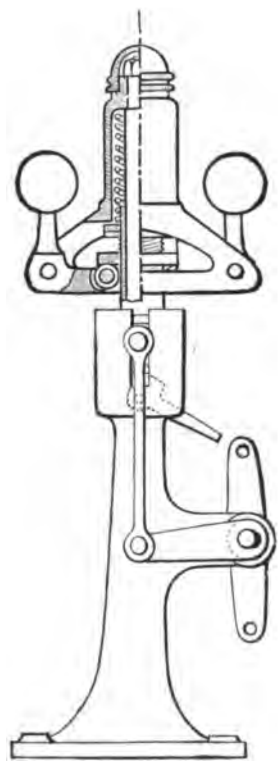


Fig. 317.

A centrifugal governor is essentially a balance, the centrifugal force of balls being opposed by a resistance, usually consisting of weights or springs; it differs in one important particular, however, in that it should not be astatic, as all delicate balances are—that is, tend to remain in any position when the balancing is effected—but should have definite positions corresponding to definite loads. As a change in speed of the engine has to occur before its governor will respond, that governor is the best which controls the fluctuations within the narrowest limits. To accomplish this result a number of conditions have been satisfied, of which the following are the most important: First, it should be

able to act upon slight variations in the speed of the engine. Second, it should be able to act quickly. Third, it should not be sensitive to rhythmical resistance induced by any throttling or by the cut-off mechanism. To satisfy the first and third conditions it is necessary that the changes due to alterations in speed should be much greater than those due to variations in resistance. For the second, all causes which retard a ready adjustment of balance between the centrifugal force and resistance should be reduced to a minimum; these are, principally, friction inertia, or momentum, of the moving parts, either of the governor or of the mechanism it may operate.

The governor with which this engine is fitted is shown in Fig. 317. It is designed to satisfy the conditions above enumerated, and it is extremely simple; the centrifugal ends of two balls situated upon the upper ends of bell crank levers are resisted by a spring at the inner ends of the same; by this mechanism the centrifugal and resisting forces can be most accurately adjusted and regulated. The speed of the governor, about 200 revolutions per minute, produces a great change of centrifugal force for a slight variation in speed; in fact, this difference varies as the square of the number of revolutions for a given fluctuation of speed, so that as compared with that of a slow running governor, say at 60 revolutions per minute, under otherwise similar conditions, its ability to be affected by changes of speed is as 100 to 9, with a corresponding insensibility to varying internal resistances. Friction also increases the difference between the maximum and the minimum speed of the engine. The use of small balls, which a high speed of governor renders possible, allows quick action, as a given force will move them through a given space quicker than it will heavier balls.

THE VALVE MOTION.

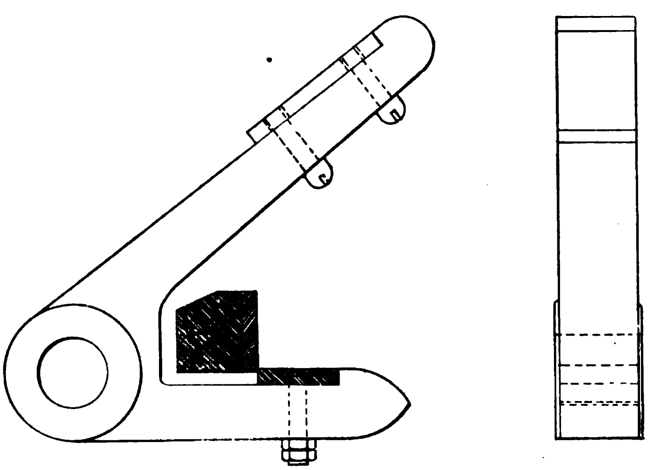


Fig. 318.

The valve motion of this engine is noticeable for its large bearings and pins. This is an important feature, for the reason that in most engines of this class these joints have been the first parts to wear loose. The strap ends are lined with phosphor-bronze bushings, which are

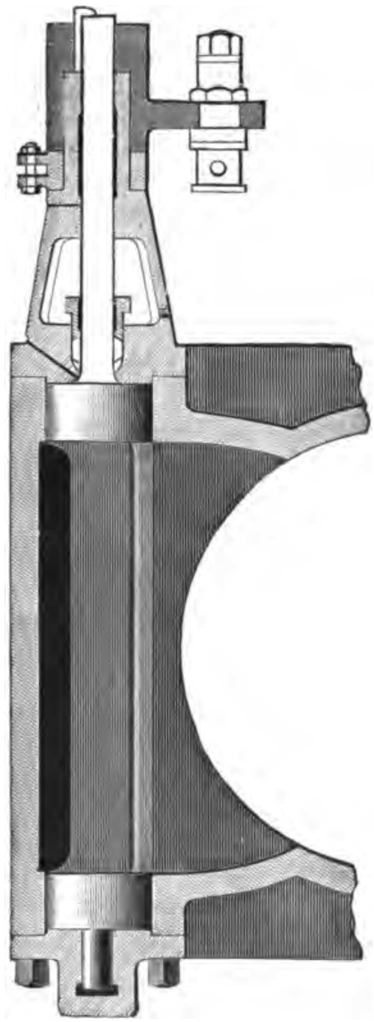


Fig. 319.

split on top, and can be closed up by set screws, the wear being taken up entirely around the circle.

Another important feature of the valve motion of this engine is the double adjustment for the steam rods. In some engines these rods are simply screwed into the strap ends, and rigidly attached at

the crab-claw end, so that the only adjustment possible is some multiple of the pitch of the screw. With the wide face crab claw, as shown in Fig. 318, there is sufficient room for a tapped hole at both ends, permitting the smallest desirable adjustment.

The illustration, shown in Fig. 319, represents the construction of the bonnet on the valve motion side, and shows a very important improvement; it extends into the recess made in the crank A, which operates the valve, thus forming a rigid bearing for the crank to swing upon. This obviates the overhung bearing, whereby excessive lateral strain is thrown on the valve stem and bearings, and rapidly wearing them out. By this construction the entire strain on the valve stem is a torsional one, which is easily transmitted. By removing the back bonnets and the keys which secure the cranks to their stems, the valves can be removed without disarranging the valve motion.

THE DASH POT.

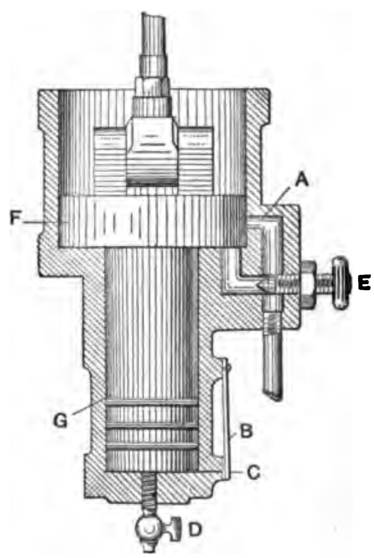


Fig. 320.

Fig. 320 represents the vacuum dash pot. The dash pot has two cylinders, a close-fitting piston F, working in the upper, and a plunger G, working in the lower cylinder. The two chambers have independent duties, exactly opposite in time and character. On the up stroke the lower cylinder affords resistance gradually by the attenuation of the air, and on the down stroke the upper cylinder produces resistance by the compression of contained air.

The perfection of the operation of the dash pot depending wholly upon the perfection of alternately created vacuum and compression,

means are provided on both cylinders to prevent the opposite conditions obtaining in their cylinders accidentally. The result then obtained is a vacuum in the lower cylinder, and the unbalanced atmospheric pressure on the area of the plunger, causing the latter to drop quickly on the release of the crab claw, and at a point in the descent the piston closes an air opening A, compressing the air remaining in the upper cylinder, bringing the parts gradually to rest.

A small check valve at C, shown on the right side of the illustration (Fig. 320), is placed near the bottom of the vacuum cylinder, and allows any air to escape which may have been taken in from leakage during the up stroke.

This valve will require some attention. The spring B should be just stiff enough to hold its leather face against the seat, with the least possible pressure, so as to not allow any air to accumulate and destroy the vacuum.

To prevent noisy contact of the piston against the bottom of the dash pot, if not previously air-cushioned, there is a leather washer secured to its under surface. In this leather there is a small portion which is shaved thin, and so cut as to form a flat valve over a small hole drilled to the piston, through which oil can work at each stroke to lubricate the vacuum plunger, which requires a certain amount of oil to make it air tight.

The drain pipe D, in the bottom of the vacuum cylinder, is used to drain away surplus oil while the engine is standing, but must always be kept closed while the engine is running.

The regulating valve E, which connects with the dash cylinder, is used to adjust the amount of air-cushioning.

The piston F should be cushioned sufficiently to prevent it from striking the bottom, and yet letting it settle without a jar. Too much cushion prevents the piston reaching its lower position until the valve rod pushes it down.

The double jump sometimes occurs when the check valve C becomes misplaced, or its spring bent, so that air can leak in and destroy the vacuum.

In starting the engine the dash pot is apt to work irregular at first, on account of the air being cold and the oil stiff. This may require opening up of the regulating valve E. But after the engine has run awhile the valve will have to be throttled to obtain the required amount of cushion.

PILLOW BLOCKS.

It will be observed that the main pillow block (Fig. 321) is practically solid, with the exception of the side boxes. These boxes are backed by two heavy wedges, having bolts extending through the cap,

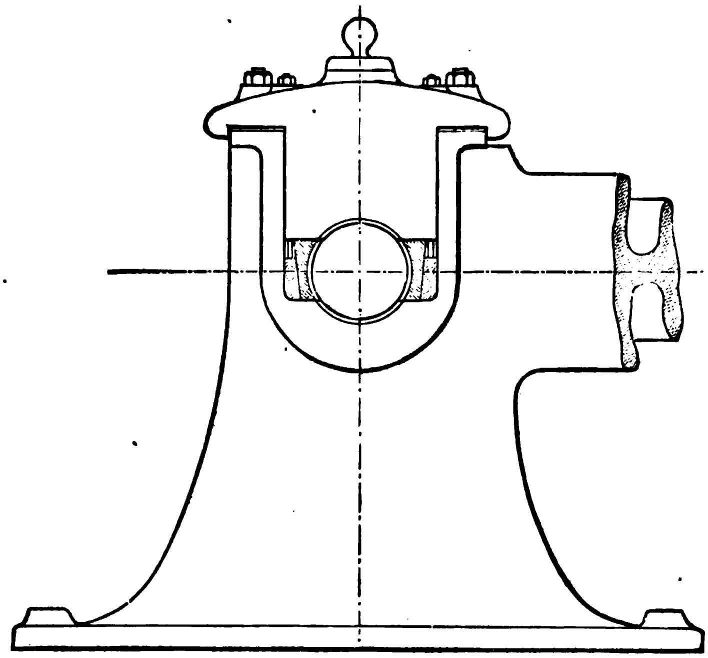


Fig. 321.

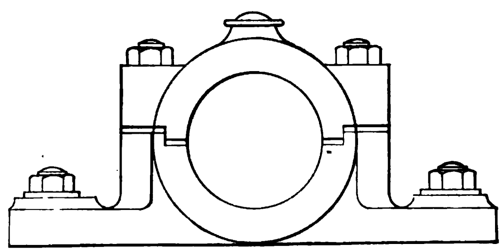


Fig. 322.

by which means the necessary adjustments are easily made. The main bearing pedestal has a large base similar to those under the slides and cylinder. The whole is substantially and conveniently arranged, and the construction is among the best found in modern practice.

COMPOUND ENGINE.

Compound engine, is a term applied to an engine which has two cylinders, arranged to use steam successively. A compound engine may consist of two entire engines, driving different loads; or two cylinders with one piston rod, acting on one crank; or two cylinders, pistons and rods coupled to cranks on the same shaft. The first application of the system was made many years ago, but without any

marked success. It was, however, generally abandoned, until 1854, when it was applied to marine purposes.

CONDENSERS.

Compound engines are generally supplied with condensers, though not necessary to the attainment of some valuable advantages. The best results, however, are attained with high pressure of steam and a condenser; the functions of which is to create a vacuum in the exhaust end of the cylinder, and thus prevent the pressure of the atmosphere from acting on that side of the piston and producing a pressure counter to the steam pressure on the opposite side. To the successful operation of a condenser a large supply of water is necessary for condensing the exhaust steam quickly. An air pump in connection with a condenser is also a necessity. A simple form of condenser with air pump is shown in Fig. 323.

This is what is termed a belted air pump and condenser. The lower part of the column is the condensing chamber; the valve shown being on the injection pipe. The air pump is vertical, single acting with double acting discharge.

The comparative advantages of the condensing engine over the non-condensing is shown in the table which follows:

FIRST-CLASS ENGINE PERFORMANCE.

A valuable table of first-class engine performance is introduced here for the benefit of the engineer and student of steam engineering, as it contains the essence of all the principles of steam as applied to work in the cylinder.

The two outside columns denote initial pressure, or the pressure at which the steam is admitted to the cylinder, and following the piston to any one of the several points of "cut off," as noted, and on a line with the number of pounds of initial pressure will be found the mean effective pressure in the columns marked M. E. P., the column marked N. C. for non-condensing, and C for condensing engines.

In the column marked "terminals" will be found the pressure of steam at the end of stroke. This pressure, however, is noted for an absolute vacuum as a base, and if the pressure on a steam gauge is wanted, subtract 14.7 from the number given in the table. Should the number in the table be less than 14.7, then there will be a vacuum in the cylinder equal to the difference between 14.7 and the tabular number.

In the M. E. P.'s the condensing column (C) shows 12 pounds greater M. E. P. than the non-condensing (N. C.). This is due to the

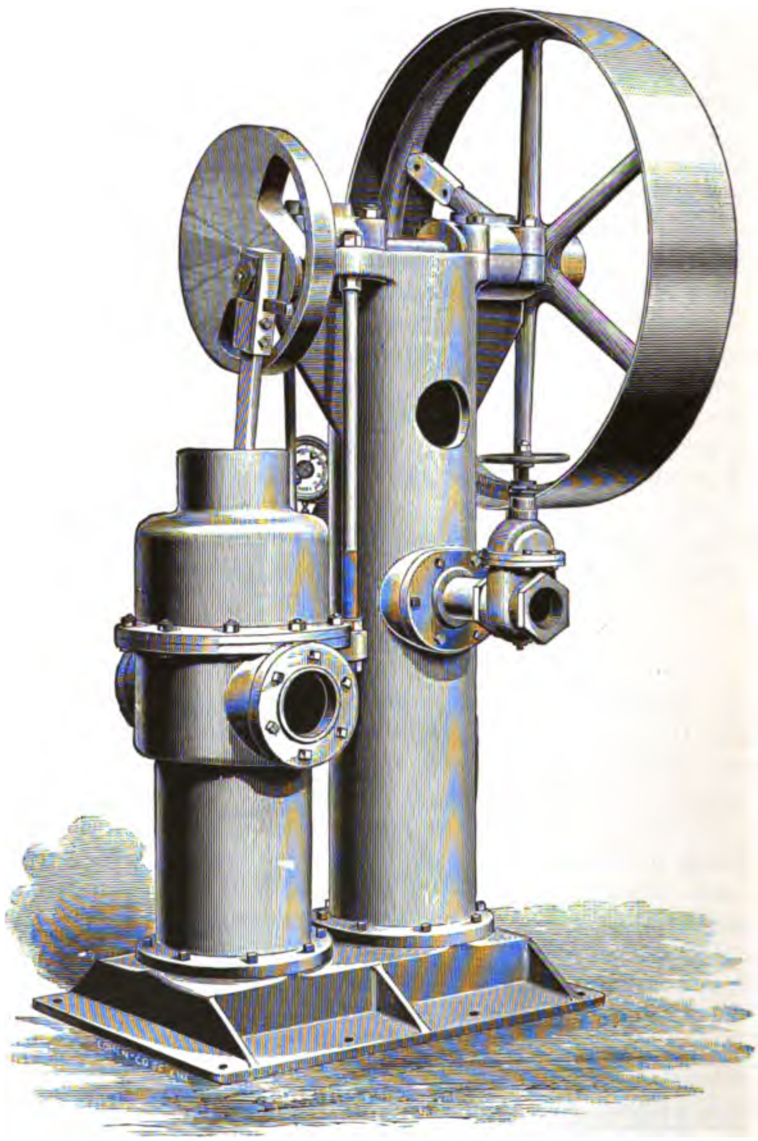


Fig. 323.

AIR PUMP AND CONDENSER.

action of the condenser, which should maintain a vacuum of 12 pounds; showing this difference in favor of the condensing engine. The four columns of water consumption denote pounds of water for each horse power for each hour of running. The two columns "theoretical," however, deal with steam as a perfect gas; a condition, by the way, that has never yet been attained, as there is always more or less water carried with the steam; hence the figures given in the "practical" columns are the very best that we can hope to attain in ordinary practice. An example of the uses of the table may be seen by taking 80 pounds initial pressure, and cut off at one-fifth of stroke, which gives 35.02 pounds mean effective pressure (N. C.), and a terminal of 20.39 pounds pressure absolute; or, in other words, 5.69 pounds by the gauge, and a water consumption of 24.3 pounds per horse power per hour, showing very good work.

The same steam in a condensing (C) engine would give 47.02 M. E. P., and a water consumption of 19.8 pounds per horse power per hour.

This table has been calculated with great care, and with a reasonable allowance for clearance, compression, etc., and represents what is possible to be done by good design, workmanship, erection and subsequent handling.

The figures relating to condensing engines are the more likely to be deviated from, as there are other things to be considered, and over which the engine has no control, but they will not be found far out of truth.

On the whole, the table sets up a standard of good work, and in practice should be approached as nearly as possible by all who own or handle steam. Tests may be made to prove the efficiency of any engine by weighing the feed water delivered to the boiler for a number of hours and dividing the whole amount by the number of hours of the test. This will give the rate per hour, which, divided by the average indicated horse power for the same time, will give the rate of water consumption. Great care must be taken that no leaks or wastes of any kind exist between the weighing apparatus and engine cylinder—no steam to be drawn from the boiler for any other purpose than to drive the engine. Even the feed pump should be a power pump and not a steam pump if driven by the same boiler, as its supply will be unduly charged to the engine.

The boiler should also be known to be in good operation, clean, and not likely to be foaming or priming. Neglect in this particular would destroy all value of the test, unless an expert calorimeter test were made to determine the percentage of entrained water in the steam.

TABLE SHOWING FIRST CLASS ENGINE PERFORMANCE.

1/10 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	3.65	15.65	6.41	69.4	20.0	51.4	14.3
45	5.42	17.42	7.00	52.0	19.9	38.5	14.2
50	7.19	19.19	7.59	43.0	19.6	31.9	14.0
55	8.96	20.96	8.17	37.9	19.2	28.1	13.7
60	10.73	22.73	8.76	33.1	19.0	25.3	13.6
65	12.50	24.50	9.35	31.4	18.8	23.3	13.4
70	14.27	26.27	9.93	29.4	18.6	21.8	13.3
75	16.04	28.04	10.52	27.8	18.5	20.6	13.2
80	17.81	29.81	11.11	26.6	18.2	19.7	13.0
85	19.58	31.58	11.70	25.6	18.1	19.0	12.9
90	21.36	33.36	12.28	24.8	17.9	18.4	12.8
95	23.13	35.13	12.87	24.2	17.8	17.9	12.7
100	24.90	36.90	13.46	23.6	17.6	17.5	12.6

15/100 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	9.05	21.05	9.07	40.7	20.5	31.3	15.2
45	11.32	23.32	9.87	36.0	20.2	27.7	15.0
50	13.59	25.59	10.72	32.9	20.0	25.3	14.8
55	15.86	27.86	11.55	30.4	19.8	23.4	14.7
60	18.12	30.12	12.38	28.7	19.7	22.1	14.6
65	20.89	32.39	13.20	27.4	19.6	21.1	14.5
70	22.66	34.66	14.03	26.4	19.3	20.3	14.3
75	24.92	36.92	14.86	25.3	19.3	19.5	14.2
80	27.19	39.19	15.69	24.4	19.0	18.8	14.1
85	29.46	41.46	16.51	23.9	18.8	18.4	13.9
90	31.72	43.72	17.34	23.4	18.6	18.0	13.8
95	33.93	45.93	18.17	22.9	18.6	17.6	13.8
100	36.26	48.26	19.00	22.0	18.4	17.3	13.6

1/5 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	13.46	25.46	11.79	34.4	21.5	27.9	16.3
45	16.15	28.15	12.87	32.6	21.1	25.7	16.0
50	18.85	30.85	13.94	30.5	20.9	24.0	15.8
55	21.54	33.54	15.00	28.8	20.6	22.7	15.6
60	24.24	36.24	16.08	27.6	20.5	21.7	15.5
65	26.93	38.93	17.15	26.5	20.3	20.9	15.4
70	29.63	41.63	18.23	25.7	20.1	20.2	15.2
75	32.32	44.32	19.31	24.9	19.9	19.6	15.1
80	35.02	47.02	20.39	24.3	19.8	19.1	15.0
85	37.71	49.71	21.46	23.7	19.7	18.7	14.9
90	40.41	52.41	22.54	23.4	19.5	18.4	14.8
95	43.10	55.10	23.62	23.0	19.4	18.1	14.8
100	45.80	57.80	24.70	22.6	19.4	17.8	14.7

1/4 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	17.34	29.34	14.49	33.7	22.0	27.2	17.2
45	20.39	32.39	15.81	31.6	21.8	25.3	17.0
50	23.45	35.45	17.13	29.8	20.6	24.0	16.9
55	26.50	38.50	18.45	28.4	21.3	22.9	16.7
60	29.56	41.56	19.77	27.3	21.2	22.0	16.6
65	32.61	44.61	21.09	26.4	21.0	21.3	16.4
70	35.67	47.67	22.41	25.8	20.8	20.8	16.3
75	38.72	50.72	23.73	25.2	20.6	20.4	16.1
80	41.78	53.78	25.05	24.8	20.4	20.0	16.0
85	44.83	56.83	26.37	24.3	20.3	19.6	15.9
90	47.89	59.89	27.69	24.0	20.3	19.3	15.8
95	50.94	62.94	29.01	23.6	20.2	19.0	15.8
100	54.00	66.00	30.33	23.1	20.1	18.7	15.7

3/10 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	20.75	32.75	17.11	32.9	22.9	27.0	18.2
45	24.13	36.13	18.67	31.1	22.8	25.5	18.1
50	27.50	39.50	20.24	29.6	22.5	24.3	17.8
55	30.87	42.87	21.80	28.4	22.2	23.3	17.6
60	34.24	46.24	23.37	27.4	22.0	22.5	17.5
65	37.61	49.61	24.94	26.7	21.9	21.9	17.4
70	40.98	52.98	26.51	26.1	21.7	21.4	17.2
75	44.35	56.35	28.07	25.6	21.5	21.0	17.0
80	47.72	59.72	29.64	25.1	21.3	20.6	16.9
85	51.09	63.09	31.20	24.6	21.2	20.2	16.8
90	54.46	66.46	32.77	24.2	21.0	19.9	16.7
95	57.83	69.83	34.33	23.9	20.9	19.6	16.6
100	61.20	73.20	35.90	23.7	20.8	19.4	16.5

35/100 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	23.70	35.70	19.80	33.0	23.9	27.5	19.3
45	27.32	39.32	21.60	31.6	23.5	26.3	18.9
50	30.94	42.94	23.42	30.4	23.3	25.3	18.8
55	34.56	46.56	25.23	29.3	23.1	24.4	18.6
60	38.18	50.18	27.04	28.3	22.8	23.6	18.4
65	41.80	53.80	28.85	27.5	22.6	22.9	18.4
70	45.42	57.42	30.66	26.8	22.5	22.3	18.2
75	49.05	61.05	32.47	26.2	22.4	21.8	18.1
80	52.68	64.68	34.28	25.7	22.3	21.4	17.9
85	56.31	68.31	36.09	25.3	22.2	21.1	17.9
90	59.94	71.94	37.90	25.0	22.2	20.8	17.8
95	63.57	75.57	39.71	24.7	22.0	20.6	17.7
100	67.20	79.20	41.52	24.5	21.9	20.4	17.6

4/10 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	26.22	38.22	22.44	33.0	24.6	28.3	20.3
45	30.08	42.08	24.49	32.7	24.3	26.9	20.1
50	33.95	45.95	26.55	30.4	24.0	25.8	19.9
55	37.81	49.81	28.60	29.2	23.8	25.0	19.7
60	41.68	53.68	30.66	28.8	23.6	24.4	19.5
65	45.54	57.54	32.71	28.2	23.4	23.9	19.3
70	49.41	61.41	34.77	27.6	23.2	23.4	19.2
75	53.27	65.27	36.82	27.1	23.0	23.0	19.0
80	57.14	69.14	38.88	26.7	22.9	22.6	18.8
85	61.00	73.00	40.93	26.1	22.8	22.2	18.8
90	64.87	76.87	42.99	25.8	22.6	21.9	18.7
95	68.73	80.73	45.04	25.5	22.5	21.6	18.6
100	72.60	84.60	47.10	25.2	22.4	21.4	18.5

1/2 CUT-OFF.

Initial Pressure	M. E. P. N. C.	M. E. P. C.	Terminals	Water Consumption Practical N.C.	Water Consumption Practical C.	Water Consumption Theoretical N.C.	Water Consumption Theoretical C.
40	30.50	42.50	27.78	32.5	26.0	28.5	22.3
45	34.75	46.75	30.33	31.5	25.8	27.6	22.1
50	39.00	51.00	32.88	30.7	25.6	26.9	21.9
55	43.25	55.25	35.43	30.0	25.4	26.3	21.7
60	47.50	59.50	37.98	29.4	25.2	25.8	21.5
65	51.75	63.75	40.52	28.8	25.0	25.3	21.3
70	56.00	68.00	43.07	28.4	24.8	24.9	21.2
75	60.25	72.25	45.61	27.9	24.7	24.5	21.0
80	64.50	76.50	48.16	27.6	24.6	24.2	21.0
85	68.75	80.75	50.70	27.2	24.5	23.9	20.9
90	73.00	85.00	53.25	27.0	24.4	23.7	20.8
95	77.25	89.25	55.79	26.8	24.2	23.5	20.7
100	81.50	93.50	58.34	26.6	24.1	23.3	20.6

ECONOMY IN THE GENERATION AND USE OF STEAM.

It is a question whether or not the term "horse power" should be applied to boilers, as it is likely to fail in giving any idea of their ability or capacity. The standard adopted by the judges at the Centennial Exposition, of 30 pounds of water per hour evaporated at 70 pounds pressure, with feed water at 100° Fahrenheit, and rating this as a "horse power," is a fair one for both engines and boilers, and has been favorably received by engineers and steam users; but as the same boiler may be made to do more or less work with less or greater degree of economy, it should also be required that the rating of a boiler be based on the amount of water it will evaporate at its most economical rate.

If all the data in connection with a boiler are known it is easy to compute the number of pounds of water it will evaporate in an hour; and if we know the water consumption per horse power per hour of the engine that is used in connection with this boiler, it is then possible to say what the horse power of this boiler is under the circumstances. If a certain boiler with one kind of setting, one grade of fuel, and excellent draft would develop, say 100 horse power, the same boiler in another setting, with another grade of fuel and poor draft, might do well if it did one-half as much work.

Take the position, that the boiler is 100 horse power, and is developing that amount of work through some ordinary slide-valve engine that uses 50 pounds of water per horse power per hour; with a good Corliss engine that is using but 25 pounds per hour, the above boiler would develop 200 horse power.

There is no trouble whatever to find engines running, some using more than 50 and others less than 25 pounds of water per horse power per hour.

A comparative record of tests, on excellent authority, showing the relative efficiency and amount of water used per horse power per hour by a number of ordinary slide-valve engines is here given:

Lbs. of water per H. P. per hour	32.34	33.65	35.52	38.83	46.35	58.67	51.00	56.09	66.81
Relative efficiency...............	.715	.681	.651	.595	.499	.494	.453.	.412	.346

The relative efficiency is compared with the attainable results of Corliss engine taken at 1000.

Thus the amount of water used per horse power per hour gives convincing evidence of the difference in engines, and is the only true basis of comparison.

The coal required per horse power per hour is evidently dependent in every case upon the economy of the engine and boiler jointly.

With an evaporation of 9.25 pounds of water per pound of coal, and thirty pounds of water per horse power in the engine, there would be required per horse power per hour 3.24 pounds of coal. Such boiler performance, however, is not common in average practice; so, generally, a low cost of power in fuel is due to using an excellent engine in connection with only a fair boiler duty. An example of this may be seen in a printed report of a test of two steam plants—we will call them "A's" and "B's."

Said circular is issued by "A," who claims the highest efficiency, and calls especial attention to his engine as using the least coal:

	A.	B.
Water used per horse power per hour..................	43.84 lbs.	36.64 lbs.
Coal used per horse power per hour....................	3.65 lbs.	4.07 lbs.
Relative efficiency...	1.000 lbs.	0.897 lbs.

Which of these engines is the more economical—the one that uses but 3.65 pounds of coal, or the one that uses but 36.64 pounds of water? We can easily see from the figures that it was the boiler that made A's engine use the less coal, and the engine appears in a bad light, as it was not near so efficient in duty as B's, when rated by the only true standard—that of water consumption. Had the engines been changed, each to the other's boilers, then both A and B would have been surprised.

Presuming an engine to be mechanically perfect—that is, tight valves, piston packing and joints—its economical performance depends on four things: A quick and full opening of the admission valve, allowing full boiler pressure to be admitted to the cylinder, an instantaneous closure of the same at the proper time, and this time to be varied, automatically, according to the varying loads and steam pressure; a late but perfect exhaust opening, and an exhaust closure early enough to secure necessary cushion for piston. The reverse of this is a sluggish opening of steam valve, with a decreased speed of movement, and at no time allowing full boiler pressure to the cylinder, an equally imperfect valve closure, and always late, with no variation whatever; an early, cramped and sluggish exhaust opening, combined with a like exhaust closure. Such will be found in the ordinary slide-valve engine.

SLIDE VALVE AND CORLISS ENGINE DIAGRAMS.

The indicator diagrams, shown in Fig. 324, one over the other, are those of a Corliss and an ordinary slide-valve engine; the dotted line the Corliss, and the solid line the slide valve. Drawn thus is graphic-

ally shown the relative performance of the two engines, cylinders same size, doing the same work, and using the same boiler pressure—say 82 pounds. In the Corliss diagram we see the steam is promptly admitted at A to nearly full boiler pressure, and that pressure continued until nearly a sufficient quantity has been taken to do the work of that stroke, then the valve is instantly closed at B, and no more steam is taken from the boiler. The steam now confined in the cylinder continues to drive the piston onward by its expansive force, increasing in volume to fill the cylinder, and the pressure reduced, as shown by the curved line B C. At the point C the piston has about completed its stroke, and the pressure has fallen to about $8\frac{1}{2}$ pounds,

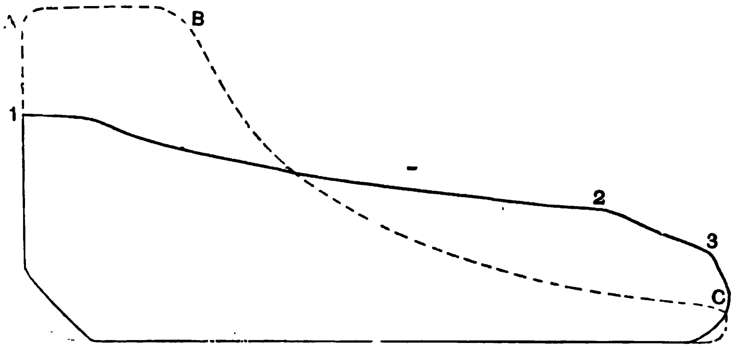


Fig. 324.

and the difference in pressure between B and C has been utilized in producing power; its vitality is expended, leaving nothing to be lost, and only a mere breath to be exhausted in the air.

For the return stroke steam is admitted to the other end of the cylinder, and the same order of things takes place. From this plan of stopping the flow, or cutting off the passage of steam from the boiler to cylinder, we get the name of "cut-off engine."

The fact must be noted, that if the steam valve should remain open too long, the speed of the engine would be increased; and if closed too soon, or before steam enough had entered the cylinder to do the work, the engine would run slower, and the speed change with every change of load or work. To control such variation a governor is used, and by its action the steam valves are closed at exactly the right time, changing as often as necessary, thus making the engine an "automatic cut-off engine."

In the ordinary slide-valve engine the action of the valve is the same under all changes of boiler pressure or load on engine, and the speed is controlled by "throttling," that is, contracting or enlarging the opening through which the steam passes on its way from the boiler to the engine. Referring again to the diagram and tracing the solid

line, we find that at the outset we are far short of getting boiler pressure at 1. This is a loss, as can be seen by comparing the height of the line at A B with that of 1 2. Thus steam is blown into the cylinder, wearing itself out, as it were, until the piston reaches 2, when the valve closes and expansion begins and continues to point 3, when of necessity the exhaust must be opened to clear the cylinder for the return stroke, and right here we have a pressure of some 25 pounds to throw away—a cylinder full of 25-pound steam thrown away! This wearing the steam out by tortuous and insufficient openings through which it must pass in order to reach a moving piston is a great loss of pressure, and is plainly shown by the diagram, as is also the loss of the use of the pressure contained in the cylinder full of steam, which is thrown away for want of opportunity to take advantage of its expansive nature. The pressure thus wasted, stroke after stroke, day after day, must be generated by the burning of fuel, and for this fuel the users of such engines are obliged to pay, and not infrequently are blind to the fact that it is thrown away.

There are other points which might be named in this connection, but it may be stated, as the result of a large number of experiments, carefully made with different engines, working under various conditions, for the purpose of comparison from a working stand point, that the Corliss engine uses an average of 25 pounds of water per horse power per hour, and the ordinary slide valve or throttling engine uses an average of 40 pounds of water per horse power per hour. Or, in other words, the Corliss engine will do as much work on 5 pounds of coal as an ordinary slide valve can do on 8 pounds.

This being the case, it naturally follows that a smaller boiler can be used to generate a given power, and make a saving in cost of outfit in this particular. Presuming that 15 square feet of heating surface in a boiler were required per horse power for a slide-valve engine, then as a Corliss engine will effect a saving of one-third, 10 square feet per horse power will be sufficient for its needs. Again, should a certain boiler be fully worked to supply a certain slide-valve engine, a Corliss could displace this engine, and the boiler would then be able to do one-half more work, and as easily as before.

FEED WATER HEATING.

In the preceding pages boiler performance has been discussed, and it was stated that a horse power consists of 30 pounds of water evaporated per hour, with the feed water delivered at a temperature of 100° Fahrenheit, and converted into steam at 75 pounds pressure.

This point should always be taken into consideration in determining the evaporation of steam boilers. It is plainly unfair to com-

pare the performance of a boiler which is fed with water at 60° Fahrenheit with one that is supplied at a temperature of 212° Fahrenheit; therefore, it naturally follows that the temperature of the water must be noted when testing a boiler, and then calculate what the amount would have been had the feed been at 212° Fahrenheit.

This computation would be a tedious one, and beyond the ability of the average engineer; but the work is very much simplified by a table, which has been prepared from the formula used in such cases.

In using the table the temperature of the feed water, the rate of coal consumption and the steam pressure in boiler, should be known. By reference to the table a factor will be obtained which, multiplied by the pounds of water evaporated per pound of coal, will give the number of pounds of water that would be evaporated had the feed water been injected into the boiler at a temperature of 212° Fahrenheit.

Thus, suppose a boiler carrying 80 pounds of steam were fed direct from a spring or well at 50° Fahrenheit, and on weighing this water as pumped to the boiler, and also the coal burned during the time of test, the boiler is found evaporating only 7½ pounds of water to one of coal. This would look wasteful, but by using the table it is found that there could have been evaporated 9 pounds had the feed been heated to 212° Fahrenheit. ($7\frac{1}{2} \times 1.204 = 9.03$.)

The use of heaters for raising the temperature of feed water is becoming general, yet there seems to be a great variety of opinions as to their use and action, when the numberless forms and claims made for each are considered. Those that make use of the exhaust steam from engine, pumps, etc., for heating are the most common, as they make use of what would otherwise go to waste, and they are highly to be commended. Two forms of these may be given notice as thoroughly practical and effective: A jet, or open heater, consisting of a vessel into which the exhaust steam is thrown; also a jet or spray of water, which in falling takes up more or less of the heat of the steam while passing through it, and is drawn off at the bottom by the pump to supply the boiler. Another form is the surface or closed heater, with a series of copper, brass, or other pipes, around which the exhaust steam circulates, while the water is forced through the pipes on its way to the boiler, and is under boiler pressure while in the heater.

The open heaters are very effective in a general way, but they must not be, in any case, expected to heat water over 208° or 210° Fahrenheit. The reason for this is that water vaporizes or boils at 212° Fahrenheit, when heated to that point in a vessel with free outlet; therefore, it would break into steam and pass out with the exhaust if heated to that point in an "open heater."

TABLE OF FACTORS OF EVAPORATION.

TEMPERATURE OF FEED.	BOILER PRESSURE IN POUNDS PER SQUARE INCH.												
	30	40	50	60	70	80	90	100	110	120	130	140	150
32 Degrees	1.207	1.211	1.214	1.217	1.220	1.223	1.225	1.227	1.228	1.231	1.233	1.234	1.236
40 "	1.199	1.203	1.206	1.209	1.212	1.214	1.217	1.219	1.221	1.223	1.224	1.226	1.228
50 "	1.188	1.192	1.196	1.199	1.201	1.204	1.206	1.208	1.210	1.212	1.214	1.216	1.217
60 "	1.178	1.182	1.185	1.188	1.191	1.194	1.196	1.198	1.200	1.202	1.204	1.205	1.207
70 "	1.167	1.171	1.175	1.178	1.181	1.183	1.185	1.188	1.190	1.191	1.193	1.195	1.196
80 "	1.157	1.161	1.165	1.168	1.170	1.173	1.175	1.177	1.179	1.181	1.183	1.185	1.186
90 "	1.147	1.151	1.154	1.157	1.160	1.162	1.165	1.167	1.169	1.171	1.172	1.174	1.176
100 "	1.136	1.140	1.144	1.147	1.150	1.152	1.151	1.156	1.158	1.160	1.162	1.164	1.165
110 "	1.126	1.130	1.133	1.136	1.139	1.142	1.144	1.146	1.148	1.150	1.152	1.153	1.155
120 "	1.116	1.120	1.123	1.126	1.129	1.131	1.134	1.136	1.138	1.140	1.141	1.143	1.145
130 "	1.105	1.109	1.113	1.116	1.118	1.121	1.123	1.125	1.127	1.129	1.131	1.133	1.134
140 "	1.095	1.099	1.102	1.105	1.108	1.110	1.113	1.115	1.117	1.119	1.120	1.122	1.124
150 "	1.085	1.088	1.092	1.095	1.098	1.100	1.102	1.104	1.106	1.108	1.110	1.112	1.113
160 "	1.074	1.078	1.081	1.084	1.087	1.090	1.092	1.094	1.096	1.098	1 100	1.101	1.103
170 "	1.064	1.067	1.071	1.074	1.077	1.079	1.081	1.084	1.086	1.087	1.089	1.091	1.092
180 "	1.053	1.057	1.060	1.064	1.066	1.069	1.071	1.073	1.075	1.077	1.079	1.080	1.082
190 "	1.043	1.047	1.050	1.053	1.056	1.058	1.061	1.063	1.065	1.067	1.068	1.070	1.072
200 "	1.032	1.036	1.040	1.043	1.045	1.048	1.050	1.052	1.054	1.056	1.058	1.059	1.061
210 "	1.022	1.026	1.029	1.032	1.035	1.037	1.040	1.042	1.044	1.046	1.047	1.049	1.051

PERCENTAGE OF SAVING IN FUEL BY USING HEATERS.

(BOILER PRESSURE 90 POUNDS.)

TEMPERATURE WHEN DELIVERED TO BOILER.	PRIMARY TEMPERATURE OF FEED WATER.							
	32°	40°	50°	60°	70°	80°	90°	100°
60 Degrees	2.3	1.7	0.8	....	....	....	....	....
70 "	3.2	2.5	1.7	0.8	....	....	....	....
80 "	4.0	3.4	2.5	1.7	0.8	....	....	....
90 "	4.9	4.2	3.4	2.5	1.7	0.8	....	....
100 "	5.7	5.0	4.2	3.4	2.5	1.7	0.8	....
110 "	6.6	5.9	5.0	4.2	3.4	2.5	1.7	0.8
120 "	7.4	6.7	5.9	5.0	4.2	3.4	2.5	1.7
130 "	8.3	7.6	6.7	5.9	5.0	4.2	3.4	2.5
140 "	9.1	8.4	7.6	6.7	5.9	5.0	4.2	3.4
150 "	9.9	9.3	8.4	7.6	6.7	5.9	5.0	4.2
160 "	10.8	10.1	9.3	8.4	7.6	6.7	5.9	5.0
170 "	11.6	11.0	10.1	9.3	8.4	7.6	6.7	5.9
180 "	12.5	11.8	11.0	10.1	9.3	8.4	7.6	6.7
190 "	13.3	12.6	11.8	11.0	10.1	9.3	8.4	7.6
200 "	14.2	13.5	12.6	11.8	11.0	10.1	9.3	8.4
210 "	15.0	14.3	13.5	12.6	11.8	11.0	10.1	9.3
220 "	15.9	15.2	14.3	13.5	12.6	11.8	11.0	10.1

The closed, or coil heater, can be used to heat water to 225° or 230° Fahrenheit, but would make a back pressure of 4 to 6 pounds on the engines, exhausting into it, and this would be an objection in most cases, yet situations exist where such back pressure is maintained for other reasons, and advantage could be taken of it. A temperature of 212° to 214° Fahrenheit can be easily attained in a closed heater without noticeable resistance to the engines, provided the heater is large enough and properly adapted to its work. It is never profitable to have a small heater of any kind.

Still another form of heater may be used which takes advantage of the lost heat in another direction, and that is by way of the stack. Great care must be used in applying such, as it may result in anything but satisfaction. Where the boiler is small and the fire has to be forced, and a very hot breeching and stack is the result, then this form may be used to some profit. The proper remedy in such cases is, by all means, a larger boiler; yet sometimes this is not expedient, and a coil of iron pipe may be placed in the stack or breeching, and the feed water forced through it to the boiler with a check valve beyond it. This should never be used unless the escaping gases are at 600° Fahrenheit, or over, and should not result in cooling down to less than 450° Fahrenheit, and should always have water passing through while the fires are strong.

The value of the increased temperature of the water fed to boilers may readily be shown, and the percentage of gain computed by using the steam and water tables to be found in the chapters on evaporative and calorimeter tests. A pound of water taken at 32° Fahrenheit, and raised to 210° Fahrenheit will have required 178 units of heat. To evaporate this pound to steam at 90 pounds will require a total of 1182 heat units. Now, 178 heat units is 15 per cent. of the 1182 required, and may be supplied by a heater using the waste from the engine, thereby saving 15 per cent. of the fuel burned.

The ultimate value of a heater using exhaust steam can not be more than 15 per cent. in economy, and that when water is taken at 32° and delivered at 212° Fahrenheit. A fair result will be when water is received at 60° and delivered at 210° Fahrenheit, making a saving of 12.6 per cent., with boiler pressure at 90 pounds.

TO PUT AN ENGINE ON THE DEAD CENTER.

There are a number of ways of putting an engine on a dead center, and several of which will be here described; and the descriptions here given are applicable to other engines as well as the Corliss.

First. Where an engine is constructed with an ordinary bed plate, and the bed plate has been truly planed, the simplest and most accurate

manner of putting the engine on the dead center, is by taking a surface gauge, placing it on the bed plate immediately in front of the center of the main shaft, and setting the point of the needle of the surface gauge to the center of the main shaft. Then move the gauge toward whichever center it is desired to place the engine, and revolve the crank on the main shaft until the center of the crank pin comes fairly to the point of the needle of the surface gauge, then the engine is exactly on the dead center.

Second. Another method is to place some stationary object close to the rim of the fly wheel or main driving wheel, in which stationary object the point of one of the legs of the compasses can be put, as shown in Fig. 325

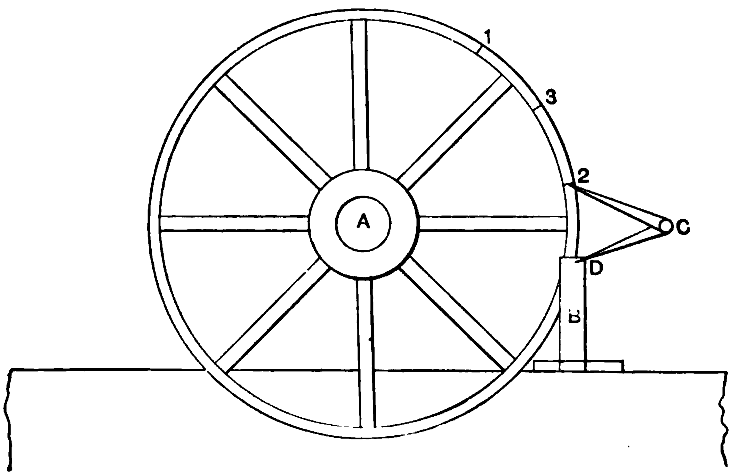


Fig. 325.

B is an improvised rest for holding the compasses while making the marks 1 and 2 on the rim wheel A. This rest is made in two parts—ordinary boards. One is nailed to the floor and B is nailed in an upright position to it.

The next step to be taken is to turn the engine toward the center on which it is desired to place it, until the cross head has traveled to within one or two inches of the end of its stroke, then stop and place a mark on the cross head and one of the guides, so that both will form a continuous line. Then place one of the points of the compasses C at D, in the board or rest B, and with the other point of the compasses scribe A an arc on the rim of the wheel, as shown in Fig. 325. The mark at 1, is the first mark; but it was made in the position mark 2 is in, as shown in the diagram, after it was made the wheel was revolved into the position shown.

After making mark 1, the compasses must neither be opened nor closed, but kept set just as they were when mark 1 was made. After making the first mark on the wheel, the wheel is again revolved until the cross head has traveled to the end of its stroke, and back until the mark on the cross head and guide come squarely line and line, and then stop. Now take the compasses and put another mark on the wheel similar to the first, but the last mark will be at 2, as shown in Fig. 325.

After the second mark has been made take another pair of compasses—as the one employed in making marks 1 and 2 on the rim of the wheel must not be disturbed, but kept in their original position—and find the center between marks 1 and 2, and put a third mark on the rim of the wheel, the third mark will be at 3, as shown in the diagram. Again place the original compasses in the original center D, in the board B, and turn the wheel A back until the upper point of the compasses is reached by mark 3; and the engine will be on the dead center. Before beginning operations to put the engine on the center, the engineer must be careful to take up all lost motion between the crank pin and piston head.

The next method is illustrated in Figs. 326 and 327.

Procure a planed board a little longer than the stroke of the engine, the lower edge of the board should be true, so that it can be leveled in position by a spirit level. Draw a line square across the center of the board at A (Fig. 326), then draw two lines B and B′ square across the board at a distance from A equal to exactly one-fourth of the length of the stroke of the engine. If the stroke of the engine is, say 4 feet, the distance from line A to line B must be 1 foot, and the distance from line A to line B′ must be 1 foot, and the two lines B and B′ will be 2 feet apart. This is done so as to get the lines B and B′ midway between the center of the main shaft and the center of the crank pin when the pin is on the center. The distance from line A to line G on either side represents the distance from the center of the main shaft to the center of the crank pin when the pin is on the center, and the distance from lines G to G represents the length of the stroke of the engine.

Next, fasten the board to the ceiling or joist overhead, in any manner most convenient, but so as to bring the face of the board E, as shown in Fig. 327, in perpendicular line with the center of the crank pin D, measuring from the face of the crank to the outer end of the crank pin. While attaching the board in position care must also be taken, by the use of a plumb line, to get the center line A on the board E (Fig. 326), in perpendicular line with the center of the main shaft. As the plumb line will hang some distance from the face of the crank, a square should be placed on the face of the crank to the center of the

main shaft, and the board E moved about until the plumb line touches the square. When the board E is in that position it should also be level. If the board is in its proper position, the line A, in Fig. 326, will be in perpendicular line with the center of the main shaft, the board will be exactly level and the face of the board will be in perpendicular line with the center of the crank pin. In short, the board will be in the position shown in Figs. 326 and 327. We are now prepared to put the engine on either dead center.

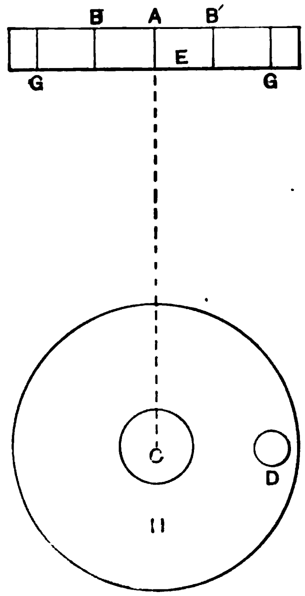


Fig. 326.

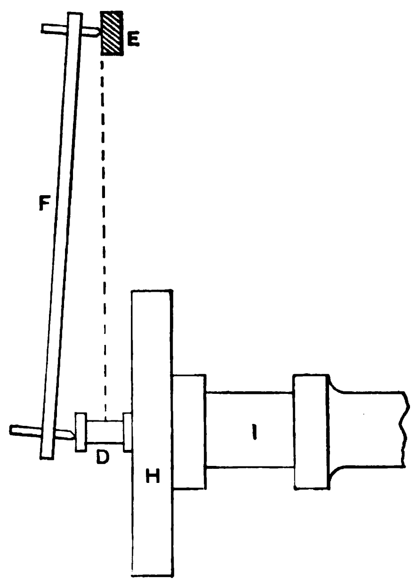


Fig. 327.

The next thing to be done is to make an improvised trammel **F**, as shown in Fig. 327, by driving two pointed nails through the board, one at each end. Then place the point of the lower nail into the center of the main shaft, and press the point of the upper nail into the board E, anywhere on line B or B′, whichever side the crank is to be placed on the center, and remove the point of the lower nail from the center of the main shaft, keeping the point of the upper nail in the trammel in the center into which it was pressed while the lower nail was in the center of the main shaft; turn the engine over until the crank pin center comes to the point of the lower nail in the trammel, as shown in Fig. 327, and the engine will be exactly on the dead center. To put the engine on the other dead center, put the point of the lower nail of the trammel in the center of the main shaft, and press the point of the upper nail of the trammel into the line B or B′,

whichever is opposite to the line first employed, and remove the point of the lower nail of the trammel from the center of the main shaft, keeping the upper nail in the center into which it was pressed; turn the engine over until the center of the crank pin comes to the point of the lower nail in the trammel, and the engine will be exactly on the dead center on that side of the main shaft.

HOW TO SET THE VALVES OF A CORLISS ENGINE.

Take off the back bonnets of the cylinder, and notice that the lines marked on the ends of the valves and on the cylinder are the lines of opening, lap and lead. The lines of lap and lead are to be followed when the engine is not to be indicated for valve adjustment.

On the back of the hub of the wrist plate will be found a center line, and a corresponding line on the hub of the trunnion which supports the wrist plate. When these two lines coincide the wrist plate will be in its central position On each side of the center line of the wrist plate trunnion will be found another line which marks the vibration of the wrist plate, and when the center of the wrist plate coincides with either of these lines the wrist plate is in its extreme position.

First. Place the wrist plate in its central position, and by means of the adjusting nuts make the lengths of the valve connections so that each steam and exhaust valve may have the proper lap. Then connect the eccentric hook to its pin on the wrist plate, and turn the eccentric loose around the shaft, and, if necessary, adjust the length of the eccentric rod so that the center mark on the wrist plate will vibrate to the extreme lines of travel marked on the trunnion.

Second. Place the crank on either dead center, and turn the eccentric on the shaft in the direction in which the engine is to run, enough more than one-quarter of a revolution ahead of the crank to show the proper opening or lead of the steam valve nearest the piston. Then tighten the set screw on the eccentric, and turn the shaft with eccentric one-half revolution in the same direction until the engine is on the other center, and notice if the lead is the same on the valve nearest to the piston as it was on the other valve; if not, adjust it by making its connection either longer or shorter as may be required.

Third. To adjust the cams for disengaging the steam-valve crab claws, let governor balls remain in their lowest position, and move the wrist plate to either extreme of its travel, and hold it in this position; then lengthen or shorten the governor rod for the steam valve at the opposite end, which will then be open so that the steel cam on the cam collar will touch the curved arm of the crab claw; move the wrist plate to the other extreme of its travel and adjust the other governor rod in the same manner. See that the adjustable brass cam on the

cam collar is placed in the proper position for preventing the hooking up of the crab claw, this being a safety stop to act if the governor ceases to rotate.

Fourth. To prove the correctness of the cut off adjustment, raise the balls of the governor to about where they would be at work, or to medium height, and block them there. Then with the eccentric connected to wrist plate, turn the engine shaft slowly in the direction in which it is intended to run, and when the crab claw is detached by the cam, measure the distance the cross head has moved from its extreme position Continue to turn the shaft in the same direction, and when the other crab claw is detached by the cam, measure the distance the crab claw has moved from this extreme position, and if the cut off is equalized these two distances will be the same. If they are not, adjust the lengths of the governor rods until the points of cut off measure alike. In the belted governor the lever on the side of the governor column carries a cam, upon which the weight of the balls rest. This will drop out of the way when the engine is up to speed, and will allow the stop motion cams to become operative and stop the engine in case the governor belt breaks.

Fifth. To start the engine, the lever on the side of the governor column must be raised so as to allow the crab claws to hook up.

Sixth. To adjust the dash pot rods, first shorten them so that the crab claw can not hook up when the wrist plate is moved from one extreme position to the other. Then with the wrist plate moved to one of its extreme travel marks, lengthen the dash-pot rod at the same end of the cylinder until the crab claw will just hook up. Then be sure to tighten the lock nut securely, as the working loose of this rod will cause the breaking of the valve bonnet. Throw the wrist plate to the other extreme mark, and adjust the other dash pot rod in the same manner.

TO ADJUST THE GOVERNOR.

If the governor "hunts" or dances, slacken the jam nuts under the balls, and screw the balls down on the arms one-half or a whole turn, then tighten the jam nuts. The engine will now run faster. Then to recover correct speed slacken up slightly on the spring. Repeat this operation until "hunting" ceases.

To decrease the speed of the engine slack off on the spring, and at the same time screw the balls further out on the arms—not too much or the governor will "hunt."

To increase the speed of the engine tighten the spring—not too much or the governor will "hunt"—at the same time screw the balls down further on the arms.

Regulations for speed can be made only to a limited extent. Engineers must therefore be careful not to go to extremes.

If the governor does not act quick enough, screw the balls further in on the arms and loosen up on the spring. If carried too far the governor will "hunt."

In making changes do not give more than one turn to the balls each time, then run the engine and observe the result.

Be careful to have the balls the same distance from the arm pins so as to evenly balance each other.

The original position for balls at stated speed for the engine is $5\frac{5}{8}$ inches from the center of balls to the center of arm pins.

"Hunting" occurs when the spring is too tight, or when the balls are too light, or when the balls are too far out on the arms for the tension of the spring.

Lack of sensitiveness occurs when the spring is too slack, or when the balls are too far down on the arms.

The best regulation takes place when the spring and length of arms are such as to just prevent "hunting"—that is, when a slight increase of either will produce "hunting."

The governor may sometimes "hunt" when starting up, but if once steadied by hand until speed is obtained, it will continue in good shape after adjustment to load, the slight change of which it will readily detect.

Beware of any tendency to tinker. Make no adjustment except after careful consideration.

CARE OF THE GOVERNOR.

A governor will be efficient only when kept in perfect condition. Its lubrication, cleanliness and proper adjustment is essential to the performance of its functions. Friction of the moving parts is the chief cause of defective regulation, and any tendency to sluggishness of action should receive immediate attention. The governor should be washed out weekly with benzine or coal oil.

REYNOLDS CORLISS ENGINE.

TO ADJUST THE VALVES OF THE ENGINE.

The working edges of the valves and ports are shown by radial lines on the ends of valves and valve chests, at the side of the cylinder opposite the wrist plate. Both steam and exhaust valves indicate lap when the lines on the valves are nearer the center of cylinder than the lines on the chests.

Fig. 329 shows wrist plate central for adjusting valve connections. Three marks on back of hub of wrist plate D, and one mark on wrist plate stand, which is bolted to the cylinder, show how eccentric connection is to be adjusted so that the wrist plate will travel correctly when in motion.

To set valves, as in Fig. 329, place the center mark on wrist plate hub even with mark on wrist plate stand, and then adjust length of valve connections so that steam valves A, and exhaust valves B, will have lap according to columns in tables; the lap being given in parts of an inch opposite the size of cylinder.

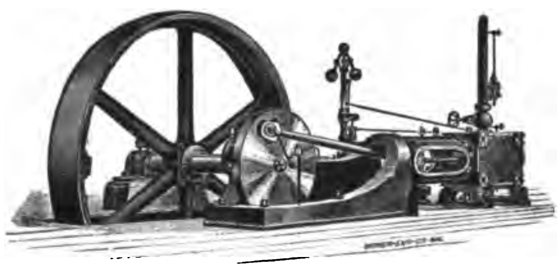


Fig. 328.

REYNOLDS CORLISS ENGINE.

Fig. 330 shows position of wrist plate D when the engine crank is on the center and eccentric set to give steam valves proper lead.

Exhaust valves will be correct if they have been set according to Fig. 329, and will need no further attention. Put crank on center and then move eccentric so that steam valves will have lead according to table; the lead being given in parts of an inch opposite the size of cylinder.

TABLE FOR SETTING VALVES.

Diameter of Cylinder.	Lap of Steam Valves.	Lap of Exhaust Valves.	Lead of Steam Valves.	Diameter of Cylinder.	Lap of Steam Valves.	Lap of Exhaust Valves.	Lead of Steam Valves.
8	$\frac{3}{16}$	$\frac{1}{32}$	$\frac{1}{32}$	24	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{64}$
10	$\frac{3}{16}$	$\frac{1}{32}$	$\frac{1}{32}$	26	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{64}$
12	$\frac{3}{16}$	$\frac{1}{32}$	$\frac{1}{32}$	28	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{64}$
14	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{1}{32}$	30	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{64}$
16	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{1}{32}$	32	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
18	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{1}{32}$	34	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
20	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{1}{32}$	36	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
22	$\frac{5}{16}$	$\frac{3}{32}$	$\frac{3}{64}$				

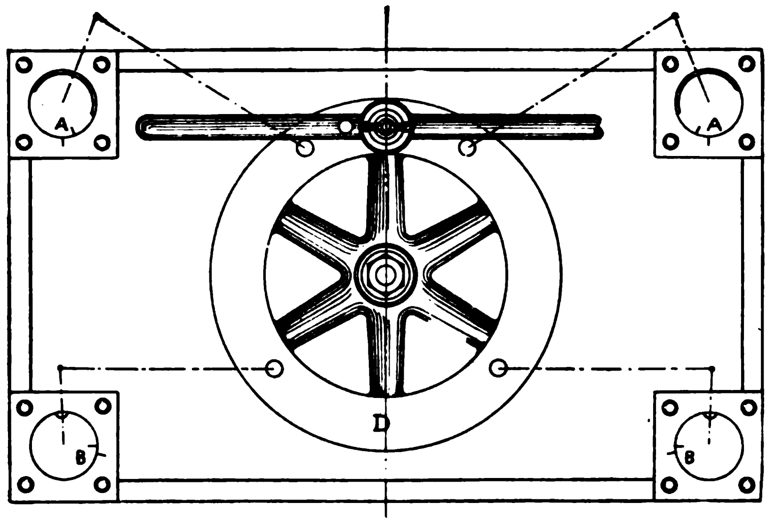


Fig. 329.

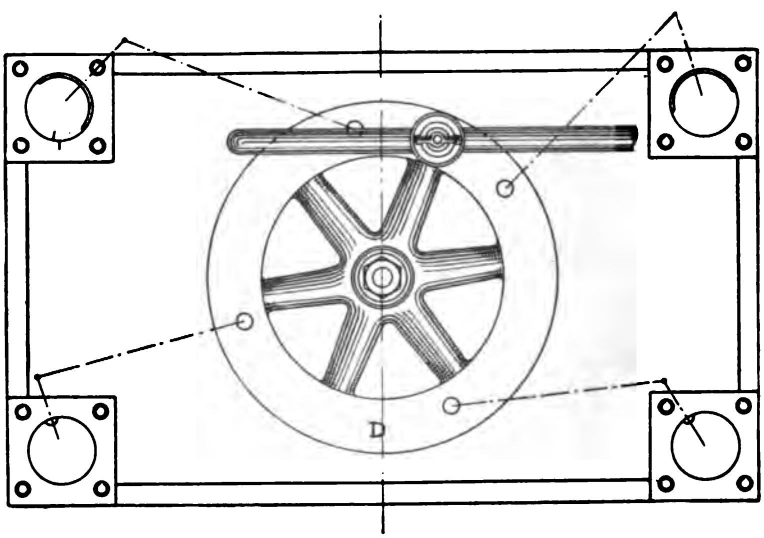


Fig. 330.

TO ADJUST THE LENGTH OF DASH-POT RODS H.

When rod is down as far as it will go, the shoulder E on brass hook should just clear the steel block F on valve arm, as shown in cut, leaving the clearance below block, as shown at G. This adjust-

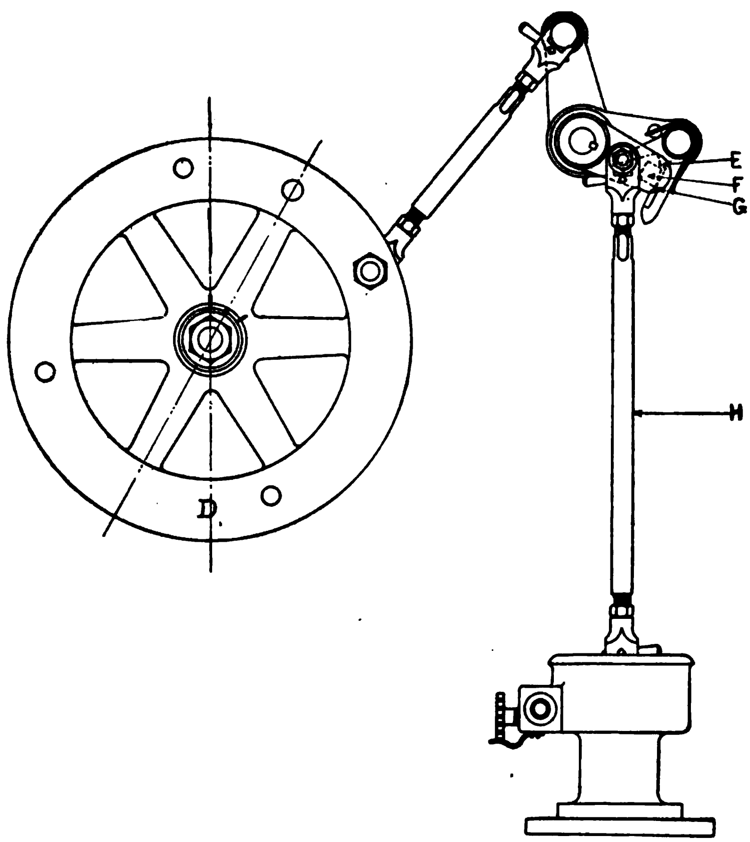


Fig. 331.

ment of rod H must be made when the wrist plate D is at its extreme throw, as shown by the mark on the back hub.

This adjustment of rod H is very important. If the rod is too short the steam valves will not open; if too long the rod will be bent or the bonnets broken, or both.

CHAPTER XX.

THE ATLAS AUTOMATIC ENGINE.

As this engine presents many novel features, its construction and the operation of the valve gear will be here explained and illustrated for the information of engineers and students of steam engineering. It is automatic in its operation of the main valve, and accomplishes the distribution of steam with a single slide valve.

In steam engines of moderate size the means almost universally employed to effect the distribution of steam is the plain slide valve. This device, on account of its simplicity, comparatively small size and little cost, and disposition to remain tight under wear, has caused it to be generally preferred to other kinds of valves. But when of dimensions suitable for large engines, or when working under high pressure, it has the disadvantage of being forced by the steam so strongly against its seat as to cause great friction, rapid wear and the absorption of a considerable amount of power to operate it. To obviate this defect many devices have been resorted to for relieving the valve of the pressure of steam, or for balancing the pressure by causing it to act equally upon opposite sides. Most of these balanced slide valves, however, have proven unsatisfactory in practice, being cumbersome, expensive, and difficult to keep in order, or requiring undesirable and costly modifications in the form of cylinder; so that the plain valve for a long time seemed likely to hold its own against them.

The advent of the single valve automatic engine, however, rendered some form of balance valve a necessity. In this style of engine the valve has not only to distribute the steam, but is made to regulate the motion of the engine, by being given a variable travel controlled by a governor. Though it is possible to make a governor powerful enough to operate an unbalanced slide valve, it can not be made to regulate very closely. The ordinary defects of the unbalanced slide valve are therefore intensified in the automatic engine, the earlier cut off and expansion of steam causing a greater difference on and under the valve. The friction causes the valve to stick at the end of its travel, and the springs or balls of the governor yielding, the ports are held open longer than the relations of load and speed require. Variations in boiler pressure also interfere with governing. Under higher pressure the engine will increase its speed, and on account of the varying resistance,

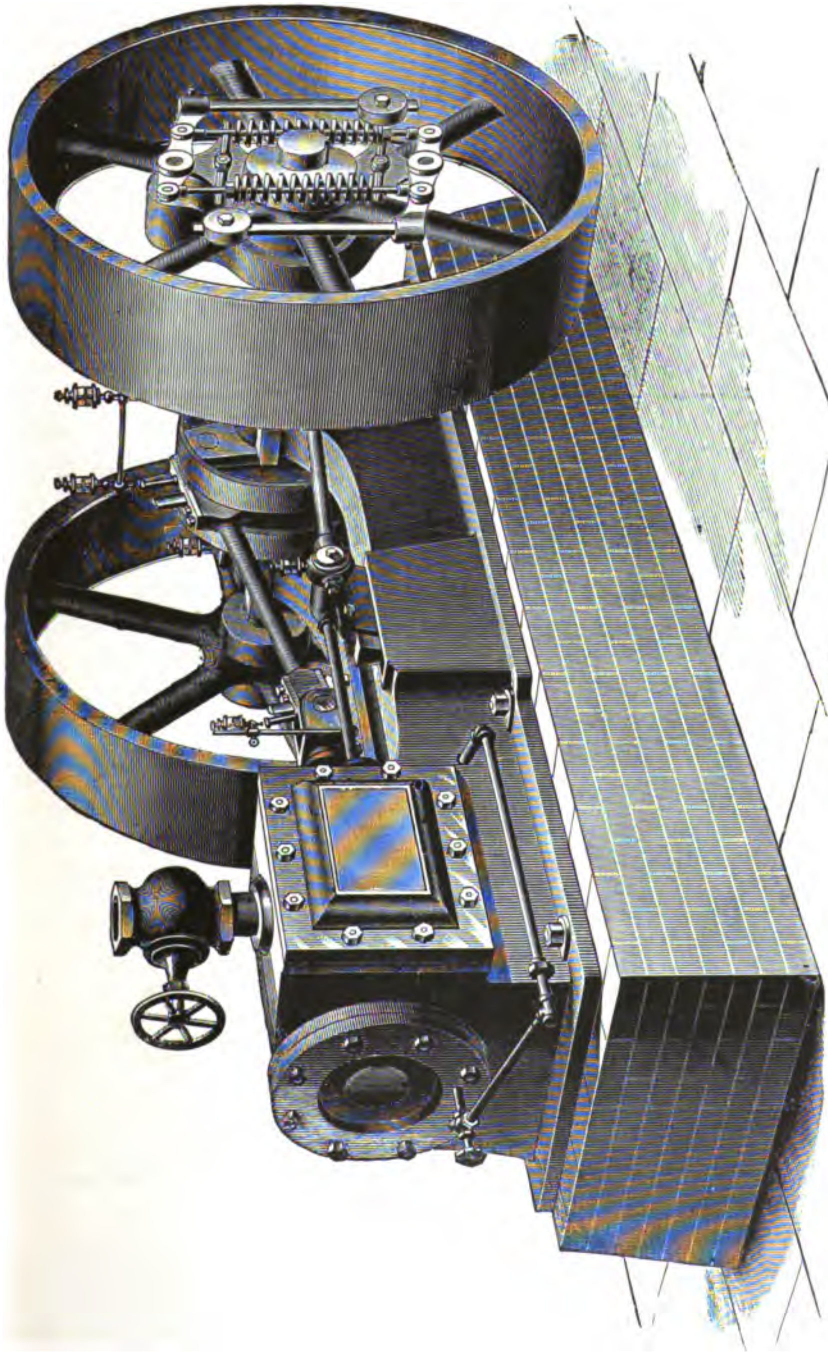
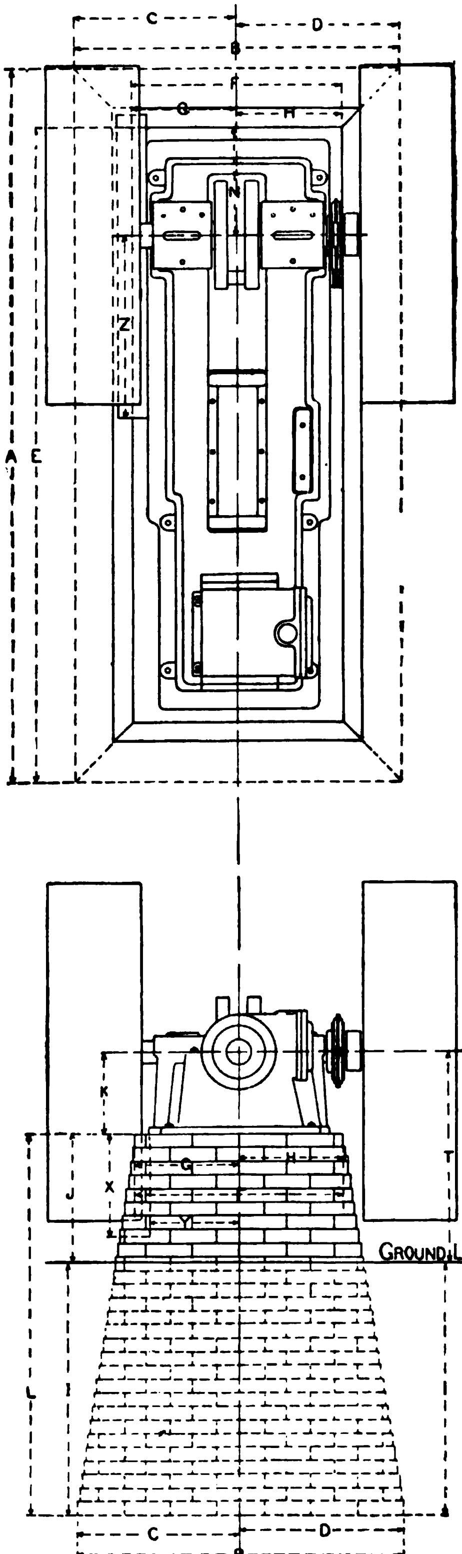


Fig. 332.
THE ATLAS AUTOMATIC ENGINE.



PLAN.

Fig. 333.

END ELEVATION.

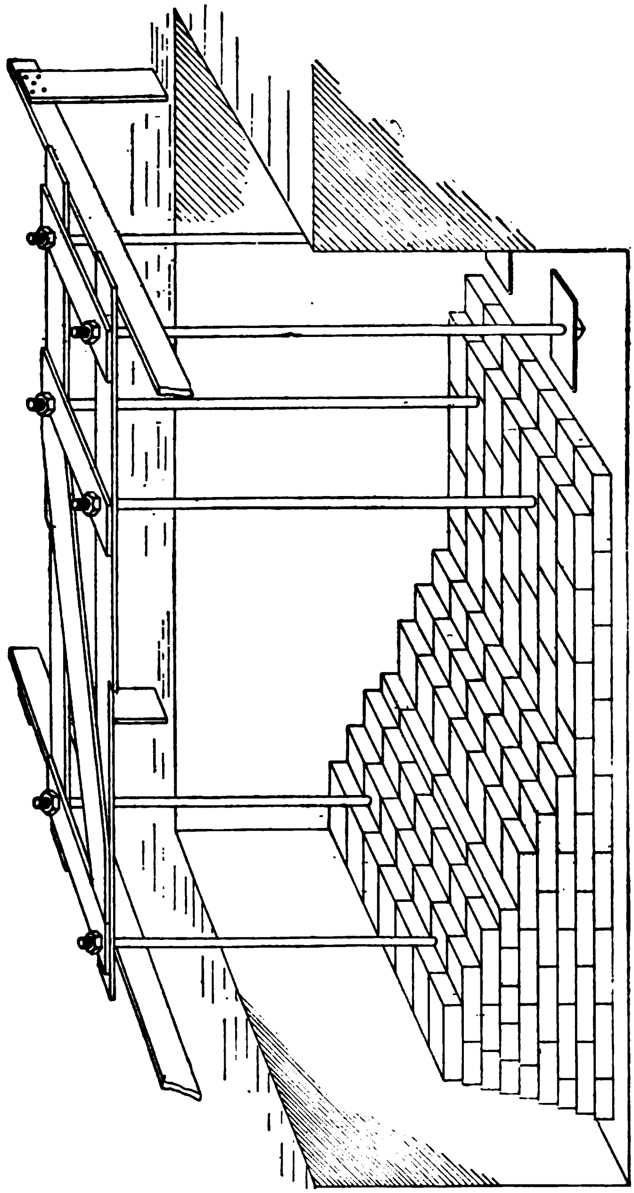


Fig. 334.
ENGINE FOUNDATION TEMPLATE.

MATERIAL FOR FOUNDATIONS.

ENGINES.	BRICKS.	SAND, Bushels.	CEMENT, Barrels.	LIME, Barrels.	SULPHUR, Pounds.
7 x 10	1,400	14	$1\frac{1}{3}$	$\frac{2}{3}$	14
8 x 10	1,400	14	$1\frac{1}{4}$	$\frac{2}{3}$	14
9 x 12	2,000	20	2	$\frac{3}{4}$	20
10 x 12	2,000	20	2	$\frac{3}{4}$	20
11 x 14	3,300	33	$3\frac{3}{8}$	$1\frac{3}{8}$	30
12 x 14	3,300	33	$3\frac{3}{8}$	$1\frac{3}{8}$	30

SPECIFICATIONS.

Dimensions of Cylinder. Diameter. Inches.	Dimensions of Cylinder. Stroke. Inches.	Horse Power at Given Revolutions per Minute, with 40 Pounds Mean Effective Pressure. Indicated Horse Power.	Standard Revolutions per Minute.	Indicated Horse Power	Revolutions per Minute.	Two Band Wheels. Diameter. Inches.	Width of Belt. Inches.	Diameters. Main Shaft. Inches.	Steam Pipe. Inches.	Exhaust Pipe. Inches.	Floor Space Required. Approximate. Length. Feet.	Length. Inch.	Width. Feet.	Width. Inch.	Shipping Weight. Pounds Approxim'te
7	10	18	240	21	275	40	7	$2\frac{15}{16}$	$1\frac{1}{2}$	2	7	0	3	8	1,900
8	10	24	240	28	275	40	7	$2\frac{15}{16}$	2	$2\frac{1}{2}$	7	0	3	8	1,900
9	12	35	225	40	260	42	9	$3\frac{7}{16}$	2	3	7	9	4	8	3,400
10	12	43	225	50	260	42	9	$3\frac{7}{16}$	$2\frac{1}{2}$	$3\frac{1}{2}$	7	9	4	8	3,400
11	14	53	200	62	230	48	12	$4\frac{1}{2}$	$2\frac{1}{2}$	$3\frac{1}{2}$	9	2	5	7	4,800
12	14	65	200	74	230	48	12	$4\frac{1}{2}$	3	4	9	2	5	7	4,800

DIMENSIONS OF FOUNDATIONS.

Engine. Size.	A. Length of Foundation at Bottom. Ft.	A. In.	B. Width of Foundation at Bottom. Ft.	B. In.	C. Center to Side of Foundation at Bottom. Ft.	C. In.	D. Center to Side of Foundation at Bottom. Ft.	D. In.	E. Length of Foundation at Top. Ft.	E. In.	F. Width of Foundation at Top. Ft.	F. In.	G. Center to Side of Foundation at Top. Inch.	H. Center to Side of Foundation at Top. Inch.	I. Bottom of Foundation to Floor Line. Ft.	I. In.	J. Floor Line to Top of Foundation. Inch.	K. Top of Foundation to Center of Cylinder. Inch.	L. Bottom to Top of Foundation. Ft.	L. In.	N. Center of Shaft to End of Top of Foundation. Inch.	T. Bottom of Foundation to Center of Shaft. Ft.	T. In.	X. Top of Foundation to Bottom of Notch. Inch.	Y. Center of Engine to Edge of Notch. Inch.	Z. Center of Shaft to Edge of Notch. Ft.	Z. In.
7 x 10	7	4	3	$6\frac{1}{2}$	1	$9\frac{1}{4}$	1	$9\frac{1}{4}$	6	0	2	$2\frac{1}{2}$	$13\frac{1}{4}$	$12\frac{1}{4}$	2	4	20	10	4	0	13	4	10	12	$11\frac{1}{4}$	1	10
8 x 10	7	4	3	$6\frac{1}{2}$	1	$9\frac{1}{4}$	1	$9\frac{1}{4}$	6	0	2	$2\frac{1}{2}$	$13\frac{1}{4}$	$13\frac{1}{4}$	2	4	20	10	4	0	13	4	10	12	$11\frac{1}{4}$	1	10
9 x 12	8	6	4	0	2	0	2	0	7	0	2	6	15	15	2	$11\frac{1}{2}$	$18\frac{1}{2}$	$11\frac{1}{2}$	4	6	15	5	$5\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{7}{8}$	2	2
10 x 12	8	6	4	0	2	0	2	0	7	0	2	6	15	15	2	$11\frac{1}{2}$	$18\frac{1}{2}$	$11\frac{1}{2}$	4	6	15	5	$5\frac{1}{2}$	$14\frac{1}{2}$	$12\frac{7}{8}$	2	2
11 x 14	10	6	4	10	2	5	2	5	8	4	2	8	16	16	3	$6\frac{1}{2}$	$17\frac{1}{2}$	$12\frac{1}{2}$	5	0	18	6	$0\frac{1}{2}$	$19\frac{1}{2}$	$13\frac{3}{4}$	2	8
12 x 14	10	6	4	10	2	5	2	5	8	4	2	8	16	16	3	$6\frac{1}{2}$	$17\frac{1}{2}$	$12\frac{1}{2}$	5	0	18	6	$0\frac{1}{2}$	$19\frac{1}{2}$	$13\frac{3}{4}$	2	8

Fig. 335.

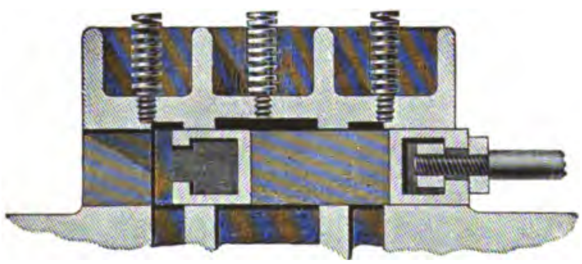


Fig. 336.

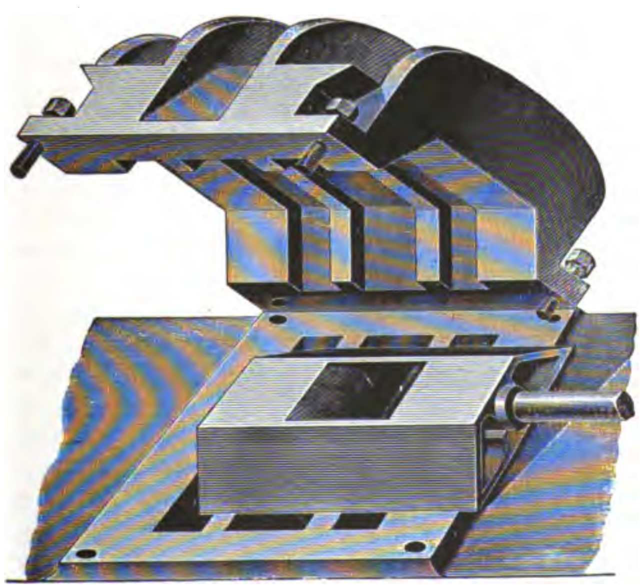


Fig. 337.

the engine will often run faster under a heavy load than under a light one. All of these defects, however, are entirely overcome in the Atlas engine, by a peculiar form of balanced valve and automatic shaft governor, all of which are exceedingly simple in their construction as well as in their operation.

The valve consists of but two principal parts, only one of which is in operation when the engine is running.

This is clearly shown in Figs. 335, 336 and 337.

Fig. 335 is a perspective view of the valve and hood complete. Fig. 336 is a section through the valve and hood on the line A B, in Fig. 335. Fig. 337 is a perspective view of the valve and seat with hood detached.

The cylinder ports and valve seat are exactly the same as for an ordinary slide valve. Upon the seat is placed a slide valve a little narrower than the length of the ports, as shown in Fig. 337, but it has a length, width of exhaust cavity, and laps, the same as an ordinary slide valve adapted to the seat. The exhaust cavity is usually of such a width as to leave no inside lap. It will also be observed that the valve is so constructed that the force of the exhaust steam is discharged against the hood instead of against the valve, as it does in the case of an ordinary slide valve. The valve is made trapezoidal in section, and finished and scraped true on all of its working surfaces. The inclined sides have an angle of about 20 degrees with the base, and the height is such as to make the end area somewhat greater than that of the steam port in the seat. The ends are cored out to secure lightness, and to obtain a uniform metal thickness throughout, so as to prevent distortion of shape by changes of temperature. The stem is attached by passing it through a lug, which is then clamped between a collar or nut on the end of the stem or valve rod.

The hood is placed over the valve to relieve it from steam pressure. This hood is carefully fitted to the valve and seat by scraping, and it is so shaped and strengthened that it will bear great pressure without distortion. Across the inside of the hood recesses are formed to correspond, in width and position, with the ports in the valve seat. The valve being slightly narrower than the length of the ports, the ends of the ports and the recesses join, thus forming passages entirely around the valve, and balancing it against whatever pressure there may be in the ports.

To prevent lateral movement of the hood, a cap screw is put through each corner into the valve seat, bottoming before it is quite screwed down, so as to leave a small space between the head of the screw and the hood, permitting the latter to rise a short distance from the seat. On the back of the hood, kept in place by studs, are a number of helical springs, which, bearing against the inside of the steam

chest cover, hold the hood to the seat. The tension of the springs is only sufficient to hold the hood under working conditions. Should over pressure occur in the cylinder, from water or otherwise, the springs will yield, and the hood immediately rising from the seat, opens communication between the cylinder and the exhaust port, and relieves the pressure.

From a superficial view it might be supposed that the making of the valve narrower than the length of the ports, would sacrifice some of the port area, but upon closer examination it will be readily seen that such is not the case. The area of the cross section of the valve being greater than the area of the port, and the recess in the hood corresponding in size and position with the port, it is obvious that when the valve has traveled so far as to uncover the width of the port the full area will be exposed, and the passage under the hood exceeding that area will cause no obstruction. Another very important effect of the mutual action of the valve and hood is at first glance not so obvious, but is readily shown. This valve opens and closes much more rapidly than is possible for a plain slide valve having the same motion upon the seat. When a plain slide valve opens, the area of the opening is only such part of the full area of the port as the width of the opening is of the full width of the port. Thus if the port be an inch wide, and $\frac{1}{24}$ of an inch be uncovered, the port will be open $\frac{1}{24}$ of its area. With the hooded valve, the recesses coinciding with the ports, the opening is not merely along the face, but around the whole perimeter of the end of the valve; and as the area of the end of the valve is made greater than the area of the port, its perimeter will be much greater than the end of the length of the port—usually about three times as great—although the width of the valve is less than the port length. Compared with the plain valve, open $\frac{1}{24}$, the hooded valve in like position would be $\frac{1}{8}$ open. This property renders this style valve peculiarly adapted for automatic governing, where, under light load, the travel of the valve is much reduced and the width of port opening very small. It combines the large port area, with rapid opening and closing, essential to economy in high speed engines. Besides it is perfectly balanced, and there is nothing to cause wear and leakage except its weight, which is very light in proportion to its size. When wear does occur, the peculiar shape of the valve renders it a very simple matter to refit it as tight as when new. Scrape off the face of the hood where it bears on the seat enough to let it down on the valve again; then scrape the valve, hood and seat to a good bearing, and it is done. The tapering sides facilitate the fitting.

All single valve automatic engines operate their valves by means of a movable eccentric, whose throw and position relative to the crank are made to vary by means of a centrifugal governor, whose action

determines the amount of port opening and time of closure necessary to maintain the required speed under the load carried. No detail of the steam engine has been the subject of more study and inventive skill than this "shaft governor." None has yet been devised better adapted to its purpose or more efficient than the one used in connection with the valve here described. In a manner peculiar to itself it combines the centrifugal principle, common to all shaft governors, with the inertia of a revolving wheel as a regulating element. This is effected without any intricacy or complication of parts. The result secured is a combination of great sensibility and power.

Fig. 338.

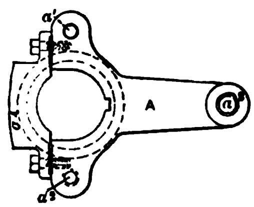
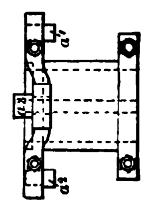


Fig. 339.



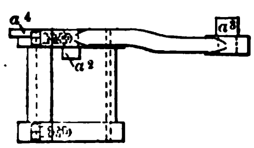


Fig. 340.

The illustrations clearly show the construction of the governor. It is composed of but seven essential parts, three of which are duplicated, making ten in all.

The hub or sleeve A (Fig. 338) is bored to fit the engine shaft, and turned on the outside. It has one long and two short arms, with wrists a^1 a^2 a^3. Opposite the long arm is a projecting lip a^4, which is finished to a radius whose center is in the wrist a^3.

The eccentric B (Fig. 341), is a flanged ring of sufficient size to surround the shaft and admit of a lateral motion sufficient to permit the amount of eccentricity required. This will be half the full travel of the valve it is required to operate, or half the sum of the widths of the steam ports and steam lap of the valve. From one side of the eccentric projects an arm like the long arm of the hub A (Fig. 338), having an eye on the end to fit over the wrist a^3 (Fig. 340). The distance from the center of the eccentric to the center of the eye, is half the steam lap of the valve shorter than the distance from the center of

the sleeve to the center of the wrist a^3. Therefore, when mounted on the wrist and engine shaft in its most central position, the eccentric will still have a throw sufficient to move the valve back and forth from line to line of the steam ports, but not to uncover them. Opposite the arm described the eccentric carries a lug piece B (Fig. 344), so formed as to embrace the lip of the hub for support and guidance, having two eyes, *i i'*, for connection with the next piece.

The dead wheel C (Figs. 348, 363 and 365), is a four-armed pulley, having its hub bored to fit loosely the turned part of the hub A (Fig. 338); its rim is provided with two lugs, *h h,'* to connect it with the

Fig. 343.

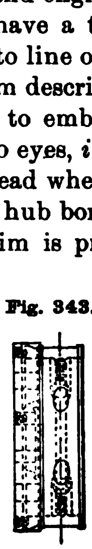
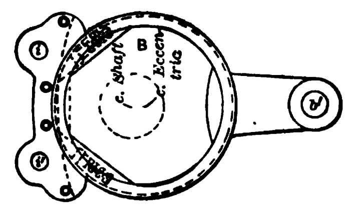


Fig. 341.



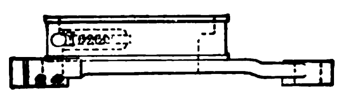


Fig. 342.

lug pieces of the eccentric and on two opposite arms, lugs and seats to receive the following pieces:

The weights D D^1 (Figs. 350, 351, 363 and 365), each having a lug *d* (Figs. 350 and 351), for attaching by means of a pin to the lugs at the ends of the arms of the dead wheel; a wrist d^1 d^1, as shown in Figs. 350 and 351, for connection with the hub A (Figs. 363 and 365), and a recess and pin d^2, as shown in Figs. 350 and 351, for attachment to the following pieces:

The springs E E^1 (Figs. 363 and 365) are helices of tempered steel; they are each provided with a connecting rod *e* (Figs. 357 and 358), and washer and nut for attaching and adjusting the tension. When in place the springs rest on the seats on the dead wheel arms, and are attached by their rods to the weights D D^1, which they tend to draw inward.

The weight links F F^1, shown in Figs. 352, 353, 363 and 365, are short bars with an eye in each end adapted to connect the hub A and the weights D D^1, by means of the wrists a^1 d^1 and a^2 d^2 (Figs. 338,

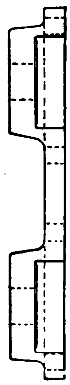


Fig. 345.

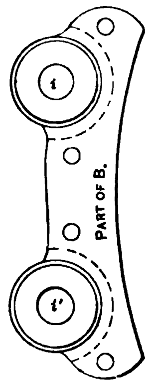
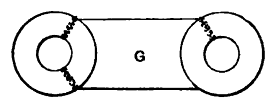


Fig. 344.

Fig. 346.



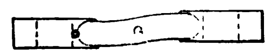


Fig 347.

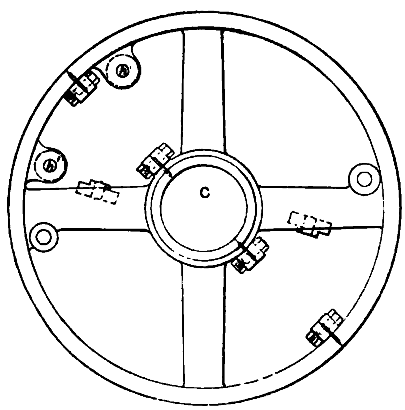


Fig. 348.

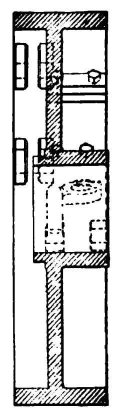
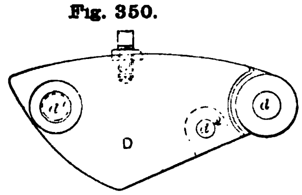


Fig. 349.

Fig. 350.



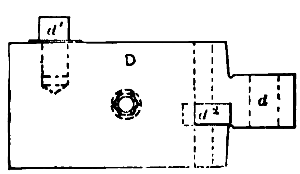
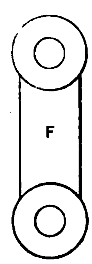


Fig. 351.

Fig 352.

Fig. 353.



339, 350 and 351.) The weights, springs and links are duplicated only for the sake of symmetry and balance. The pairs act as one.

The eccentric link G (Figs. 346, 347, 363 and 365) is like the weight link in appearance; it connects the eccentric B and the wheel C, by means of the lugs h^1 i^1 (Fig. 363), or h (Fig. 365.)

The parts already described are shown assembled in Figs. 363 and 365. They are so connected together that the joints may work freely. Wear may be taken up where necessary, and means are provided for lubricating.

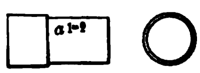


Fig. 354.

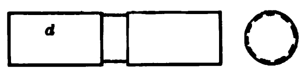


Fig. 359

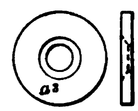


Fig. 362.

Fig. 355.

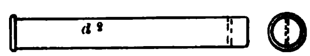


Fig. 360.

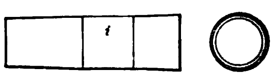


Fig. 356.

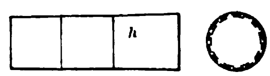
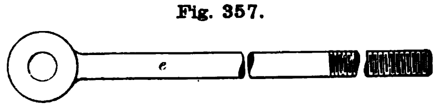


Fig. 361.

Fig. 357.



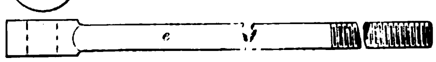


Fig. 358.

When mounted upon the engine shaft, the eccentric is placed in line with the valve-slide wrist to which it is to be connected by means of the usual straps and rod, and the hub A is keyed to the shaft in such position that the wrist a^3 will lie in the same plane as the crank pin on the opposite side of the shaft. It will be observed that the weights D D^1 and their links F F^1 form knee or "toggle" joints, connecting the hub A and wheel C, so that when the weights swing outward or inward, the wheel is compelled to turn slightly around the hub. Or conversely, turning the wheel through a small angle, will swing the weights out or in, according to the direction of the rotation. The wheel being connected to the eccentric by means of the link G,

its rotation about the shaft causes the center of the eccentric to be carried across the shaft, so as to diminish its eccentricity as the weights move outward, or increase it as they move inward. These parts are capable of two relative positions. The first is shown in Fig. 363, which is right for a left-hand engine running over. In this position, if the wheel be turned to the right, the weights will be drawn towards the center of the wheel; if turned to the left, the weights will be carried outward until the set screws outside of the weights come in contact with the wheel rim, stopping further motion. But if the set screws be removed the wheel may be turned further to the left until the weight link passes a radial position, when, the motion continuing, the weight will be again drawn inward, finally reaching the position

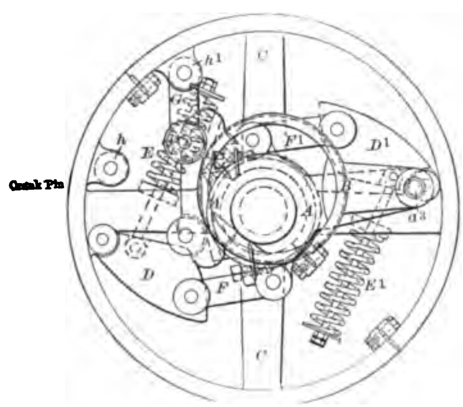


Fig. 363.

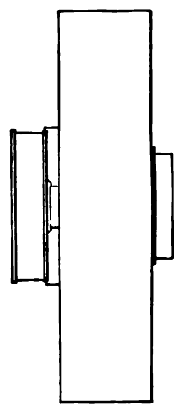
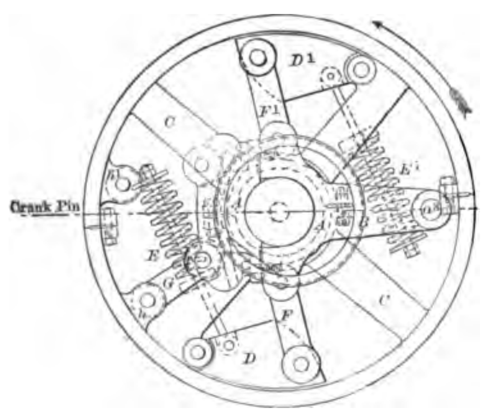


Fig. 364.

shown in Fig. 363. This, however, can nôt be done without removing the eccentric link connecting lugs *h i* (Fig. 363). That having been done, the parts may be brought to the position shown, and the link replaced so as to connect the other pair of lugs, h^1 i^1. This reverses the governor, and the engine will now run under.

The springs are of such strength, and are given such initial tensions as will just balance the centrifugal force developed in their inner position, and revolving at the number of revolutions per minute the engine is intended to run; and so proportioned that an additional revolution per minute is required to increase the force of the weights sufficiently to carry them to their outer position. Thus a variation of one revolution per minute would be sufficient to readjust the position of the eccentric from that required for the full valve opening, or full load, to that for the least opening, or no load. In this respect the weights and springs act in this governor precisely as they do in most

other shaft governors. But in this the weights can not move except by dragging themselves around the dead wheel. If they move at moderate speed, the wheel being in perfect balance and the friction inconsiderable, little resistance is offered, but the inertia of the wheel will prevent any sudden or violent movement. Other governors, not having this feature, are prone to "kick" at times with dangerous violence, a fault their makers attempt to correct by the application of friction plates or cataracts, either of which has the effect of making the governor sluggish in its action. Another very important advantage is secured from the inertia of this dead wheel: By reason of inertia a body in motion offers a resistance proportioned to its weight to any increase or diminution of its motion. Thus a wheel running loose on



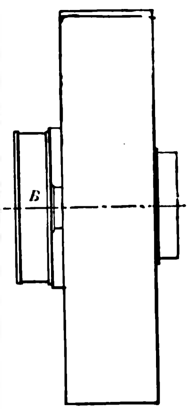


Fig. 365. **Fig. 366.**

a revolving shaft tends to revolve at the same speed as the shaft, so long as that speed is uniform, but if it change, the wheel resisting a change of speed, will not immediately take up an increase of speed in the shaft but will fall behind. If the shaft suffer a fall in speed, the wheel tending to uniform motion will overrun it. In the governor these influences affect the dead wheel, which is here so connected that action set up by a change of speed is made to adjust the admission of steam to meet and correct that change. The dead wheel, therefore, acts to reinforce the action of either of the weights, or the springs, as occasion requires. The latter may therefore be made much lighter and smaller than usual in other governors. In fact the dead wheel may be regarded as the real governing agent, the centrifugal part of the governor merely indicating the speed to be maintained To show the power with which the dead wheel can act to move the eccentric and valve, let it be assumed, for example, that the weight of the rim is 500 pounds, and

the rim is 36 inches in diameter, and that the standard speed is to be 200 revolutions per minute. Under such conditions the energy stored in the revolving rim would be 7670.6 foot pounds. If a resistance were thrown on the engine sufficient to reduce the speed to 199 revolutions per minute, at the latter speed the stored energy would be 7594.1 foot pounds, a difference of 76.5 foot pounds. The eccentric corresponding to that size of wheel requires a movement of its center of but two inches to carry the action of the valve through its entire range. Therefore, the force available for this, developed by a change of rate of one revolution per minute, or one-half of the per cent., amounts to a pressure of 459 pounds. The great energy with which

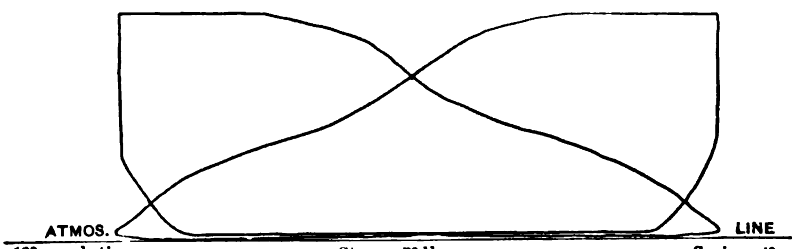


Fig. 367.

the wheel tends to maintain its speed, is of course available likewise to overcome any resistance to motion of the valve or rod. Hence the sensitiveness of this governor to variations of load and its ample power to meet them.

TO SET THE VALVES OF THE ENGINE.

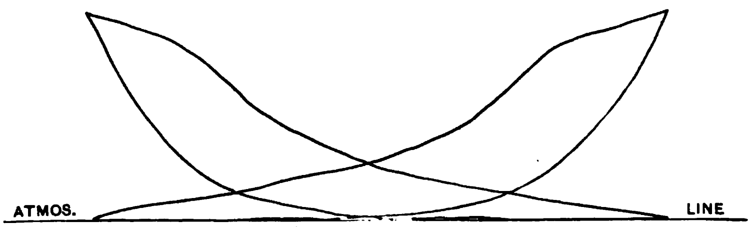
First. Turn the engine in the direction in which it is to run, until the center of the crank pin comes to the line joining the centers of the shaft and cylinder; that is, place it on the inner center.

Second. Observe that the center of the pin in the long arm of the hub on which the eccentric swings, is on the same center line extended beyond the shaft. It is obvious that if the center of the eccentric is brought to the same line, the eccentric will be in position of least throw.

Third. Mark the position of the governor spring seats, so they can be returned to the same place; then remove the springs from the governor, being careful that the shaft does not turn, and roll the dead wheel backward until the center of the eccentric comes to the line as described. The weights will then be out near the dead wheel rim. Block them in this position so there can be no movement.

Fourth. The steam chest cover having been removed, adjust the valve stem by means of nuts at the valve cross head, so that the valve will lap at the outer steam port $\frac{1}{32}$ of an inch.

Fifth. Without moving the blocking from the governor, turn the engine in the direction in which it is to run—to the outer center—and observe that the lap of the valve over the inner steam port is the same as it was on the outer steam port; if it is not, adjust the length of the valve stem and the position of the governor until the lap of ports is equalized. Then turn the crank, being on either center, unblock the weights and allow them to come in until the end of the valve comes "line to line" with the port, and turn out the stop screws in the back



120 revolutions. Steam, 62 lbs. Spring, 40.

Fig. 368.

of the weight until they prevent the weights from going out any further by coming in contact with the dead wheel rim. This completes the valve setting, and the steam chest cover may now be put on.

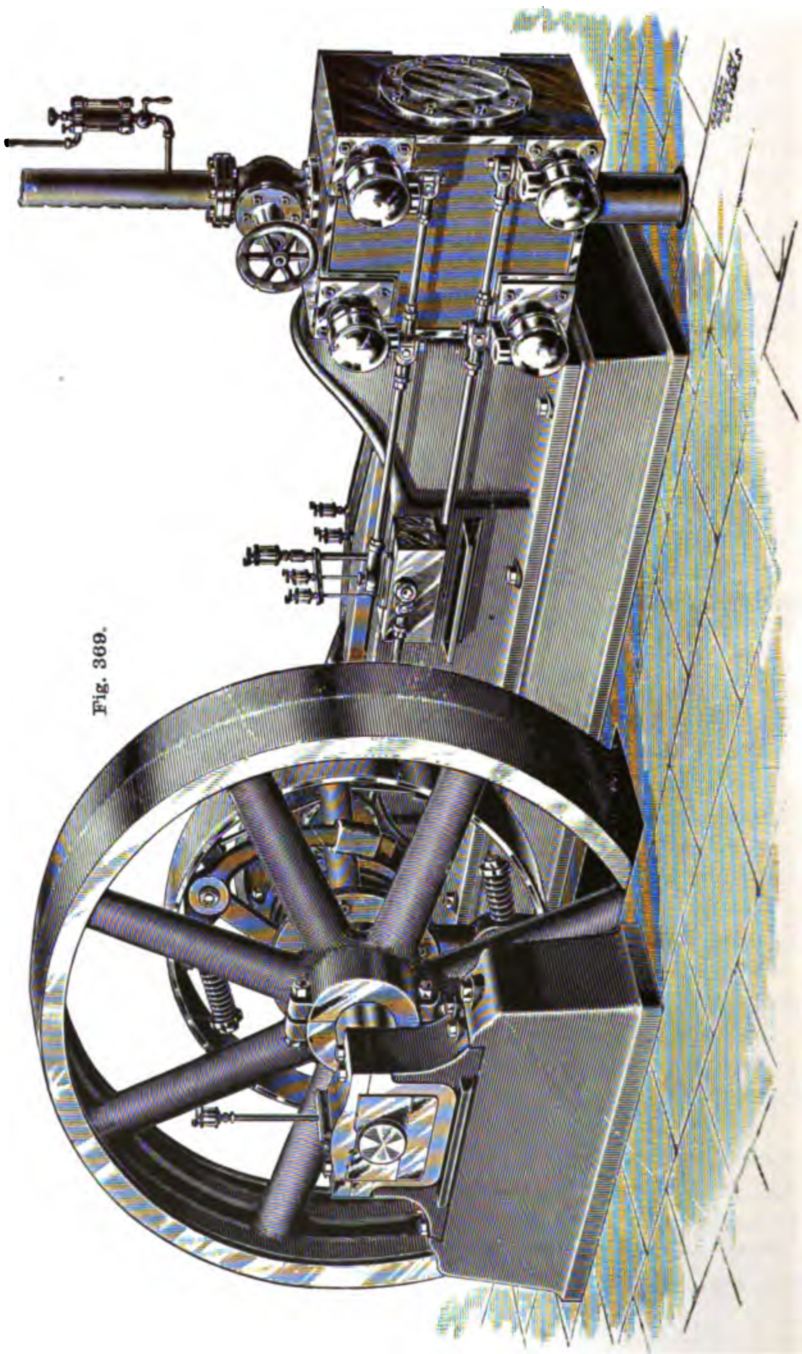
Sixth. Roll the dead wheel forward until the weights come in as far as they will, and replace the springs, adjusting them to the marks, so that they will have the same tension as before.

The speed of the engine may be varied to a limited extent by increasing the tension of the springs for greater, and diminishing it for less speed. But the springs must never be compressed so much that the spires will come together and prevent the weights from going out, and the tension must always be sufficient to overcome all friction and give the weights a lively return when they have been thrown out.

The keyway for securing the governor hub is on the same side of the shaft as the long arm. The key should be fitted sidewise, but should bear lightly at the top and bottom, otherwise the sleeve may be sprung out and prevent the wheel from turning.

Figs. 367 and 368 are fair average samples of indicator diagrams taken from this engine.

Fig. 369.



CYCLOIDAL ENGINE.

THE CYCLOIDAL ENGINE.

This engine is provided with four valves—two steam and two exhaust valves—of the multi-ported type. They have a very short travel, open and close quickly, and the steam valves may be adjusted for any port opening by a very little motion of a simple cycloid. The steam valves are controlled by a shaft governor. The exhaust valves are operated by a fixed eccentric. The steam valves are opened by a cycloid, but they are closed by steam pressure. The mechanism by which these valves are operated is plainly shown in Figs. 370 and 371. Fig. 372 is a cross section through the valves and cylinder, and shows the valves and mechanism by which they are operated. Fig. 373 is perspective view of the compound cycloidal engine. Figs. 374 and 375 show the construction of the valves, and Fig. 376 a longitudinal section of the compound engine.

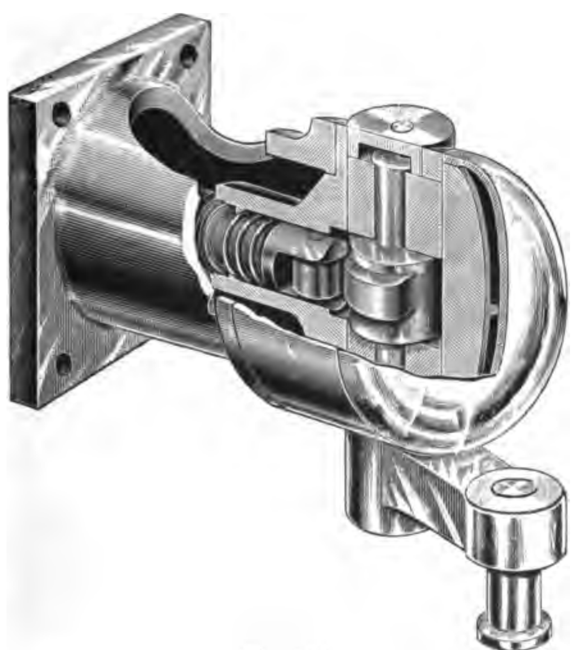


Fig. 370.

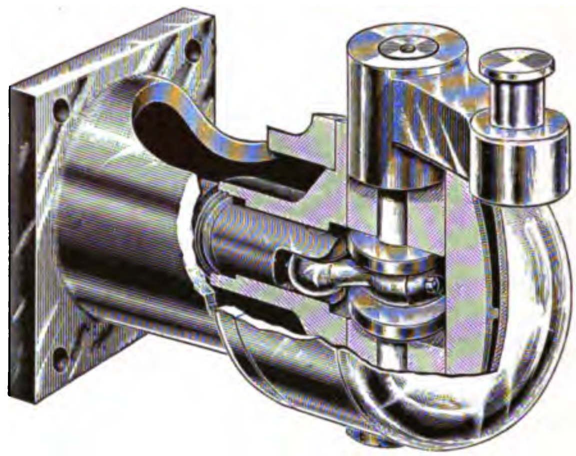


Fig. 371.

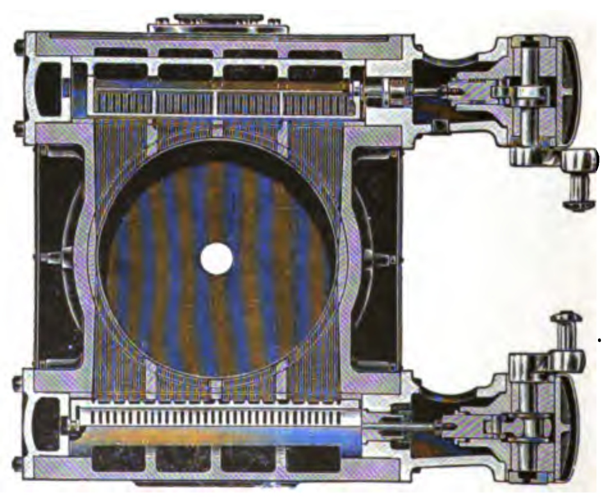


Fig. 372.

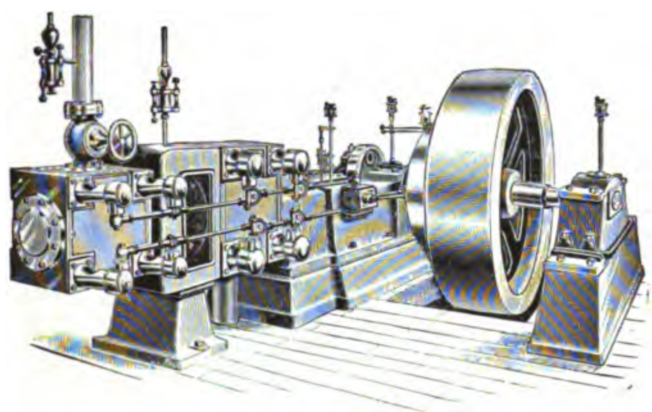


Fig. 373.
PERSPECTIVE VIEW OF COMPOUND ENGINE.

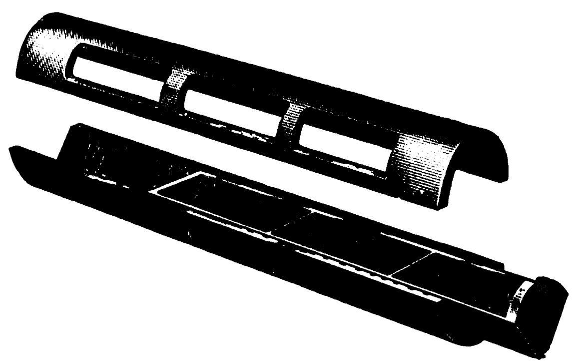


Fig. 374.

Fig. 375.

Fig. 376.
SECTIONAL VIEW OF COMPOUND ENGINE.

CHAPTER XXI.

THE BROWN ENGINE.

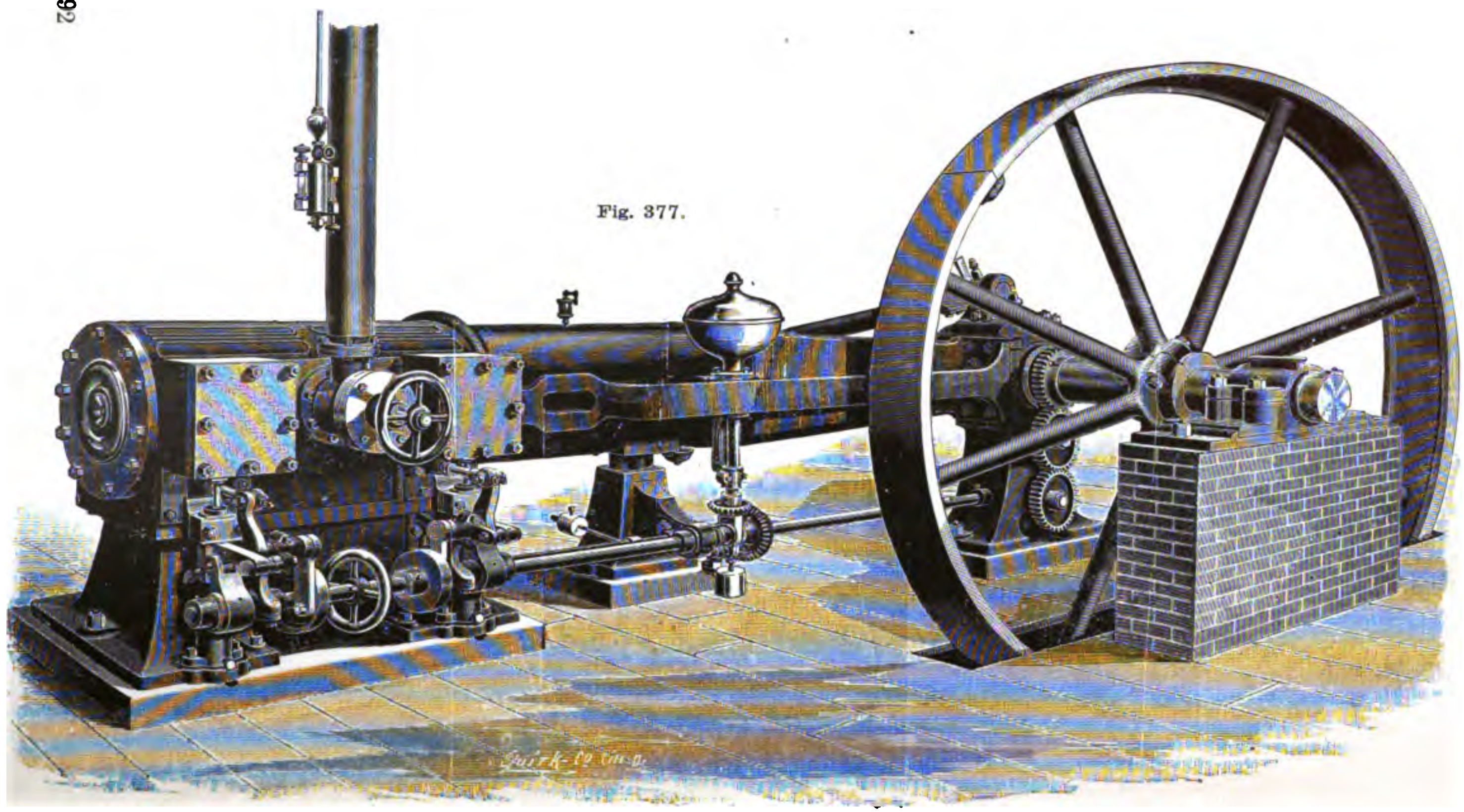
The Brown engine is one of the leading American steam engines, with features peculiar to itself, and may, therefore, properly be classed among those which distinctly differ from all other steam engines. As many of these engines are in use, in this and other countries, it is of the utmost importance to engineers and students of steam engineering that they should thoroughly understand the anatomy of this important combination of American mechanism. In order then that the information so essential to engineers in the prosecution of their profession may be imparted in its simplest form, a plain and simple description will be here given of the important details of this engine, using for the purpose of convenient and intelligent illustration, the Greenwald Brown Engine. The engineer who will study the anatomy of this engine will obtain accurate information, not only as to the details and principles of construction of this particular engine, but as to all engines of this class, no matter by whom built.

The valves of the Brown engine are plane plates, sliding upon plane surfaces, presenting opposing surfaces so distributed that no shoulders are worn thereon by the variable travel of the cut-off valves. The closure of these valves is made instantaneous and positive by the action of the steam pressure upon the area of the valve stems, the shock of the closing movement being noiselessly received, after ports are closed, by air cushions provided for the purpose. This arrangement secures positive action of the cut-off valves at any desired rate of piston speed. As the steam pressure within the steam chests and the cylinder is identical at the time of disengagement, there is no friction of the valve against the seat to retard its proper closure.

Each valve has several ports or openings, a suitable number being provided to insure prompt and ample area to maintain the closest approximation to the boiler pressure within the cylinder, up to the point of cut off, in case of the steam valves; and in case of the exhaust valves, the openings in them afford the greatest possible freedom from back or counter pressure on the exhaust side of the piston.

Each valve is entirely independent of the others, both in regard to actuation and adjustment; hence the most precise adjustment possible in modern automatic cut-off engines can be made.

Fig. 377.



THE GREENWALD BROWN ENGINE.

The valves are actuated by a cam shaft common to this class of engines, which shaft has a coincident rotary motion with the main crank shaft. The steam valves are moved by separate eccentrics having sufficient angular advance to insure a quick opening movement. The exhaust valves derive their movement from separate wall cams, the peculiar shape of which give to the exhaust valve ports a very quick opening on the exhaust side of the piston, the valves remaining at rest during the greater part of the piston's stroke, and closing quickly just as the stroke is completed, remaining so with the valve stationary during the following steam stroke of the piston.

In order to convey a clear idea of the valve mechanism, reference will be had to Figs. 378, 379, 380, 381, 382 and 383:

Fig. 378 is a side view, with the covers of the steam chest removed, and showing the side shaft, valve rigging and cut-off shaft.

Fig. 379 is a sectional plan, showing the steam chest, exhaust chamber under the cylinder, side shaft, and coupling clutch, gears for governor, and main shaft attachment.

Fig. 380 is a sectional end elevation, showing the steam chest and exhaust chamber, with the valves and their connections.

Fig. 381, shows a general arrangement from valves to main shaft

Upon reference to the figures it will be noticed that there are four valves—one steam and one exhaust valve at each end of the cylinder.

The steam valve works vertically, and in Fig. 380 is shown open admitting steam. The exhaust valve in the chamber under the cylinder moves horizontally, and is shown closed. The valve rods are connected to the valves by means of brass flanged nuts; and in case of removal of any valve or its attachments at any time, and to verify the lengths of the valve rods, a gauge *e g* and *s g* is provided for a guide, as shown in Figs. 382 and 383, the short side X being for that purpose.

To test the length of the steam valve rod, the rod is screwed into the nut and the valve pushed up in the chest as far as it will go, and the length X (Fig. 383), from the chest to the bottom of the drip cup Q, of the gauge *s g*, is the correct length. For the exhaust, the valve is also pushed in as far as it will go, and the rod screwed in; and the length X (Fig. 382) of the gauge *e g*, from the cap of the exhaust chamber to the center of the strap Z, is the proper length of the rod. The full length of the gauges *e g* and *s g* gives the position of the edge of the valves and edge of ports, when opening, measuring from the points named.

In referring to Figs. 379 and 381 it will be observed that the side shaft is driven from the main shaft by a train of gears. The gear shaft R (Fig. 379) carries a miter wheel, which engages with its mate

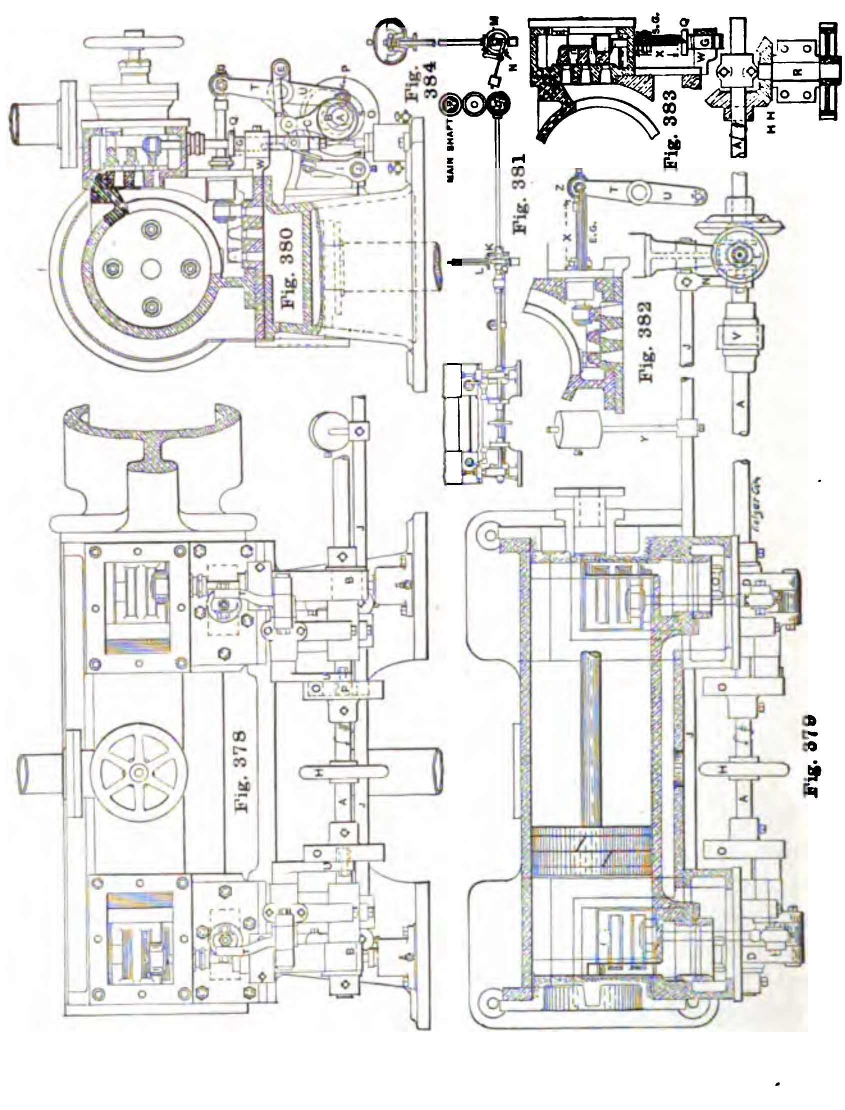


Fig. 378 Fig. 379 Fig. 380 Fig. 381 Fig. 382 Fig. 383 Fig. 384

on the side shaft running the entire length of the engine, and on which is secured the eccentrics and cams to move the valves. In the arrangement shown, the engine is adjusted to run under on the outward stroke. To run the reverse way, or over on the outward stroke, it is necessary to change the position of the miter wheel on the side shaft A to the outer end of the shaft, just opposite to the position shown in the engraving; for it must be borne in mind that the side shaft must always run in the direction indicated by the arrows. This shaft also has a bevel wheel K, which gears with a pinion on the governor spindle. Near the governor spindle, on the shaft A, is the clutch coupling V, which is so constructed that the cylinder end of the shaft A may be disengaged from the crank end while the engine is at rest, and revolves by turning the hand wheel H in the direction shown by the arrows. By this arrangement the valve rigging can be moved for the purpose of making adjustments, or for warming the cylinder.

In Fig. 380, A is the side shaft; B, steam eccentric, having arm C secured to lever D, which vibrates on E as the shaft A revolves. As the lever D is raised by the movement of the eccentric B, it engages with a steel catch on the dog F. This dog is hung in the center of the bridle G by a steel pin, and is free to move. When the lever D engages the dog F it lifts the bridle G through the dog, and therefore in opening moves upward in the guide W, carrying the valve rod and valve attached to the top of the bridle G. The upward movement of the valve will continue until the tail of the dog F comes in contact with the tripping disc and the arm I, when the dog is tripped, and the valve having no support, falls and closes.

The tripping disc is attached to the upper end of the cut-off arm I, and is readily adjusted by means of a screw and jam nut, as shown in Fig. 380. The arm I is set-screwed to the cut-off shaft J, which shaft is oscillated by the governor.

Motion is imparted to the governor through the bevel wheels K and L, as shown in Fig. 379 and 381. The governor spindle is hollow and carries the weights or balls on short arms, connected to the governor rod running down through the spindle and attached at the lower end of the bridle M, as shown in Fig. 384, and the bridle M is secured to arm N, which arm is set-screwed to cut-off shaft J. By the movement of the governor balls the rod is made to move up and down, thus imparting an oscillating motion to the arm N, cut-off shaft J, and arm I, with its tripping disc. When the tendency of the governor is to speed up, the arm I moves from the cylinder towards the dog F, thus cutting off earlier; and when the tendency of the governor is to slow down, the arm I moves toward the cylinder and from the dog F, thus cutting off later.

The exhaust valves are operated by the cams O O on the side shaft A. The pin P, with its steel roller moving in the groove S (Fig. 380), causes the arms T and U to vibrate back and forth, giving motion to the valves. Portions of this groove are concentric with shaft A. In this part of its path the valves are motionless; but during the other part of the revolution of the side shaft the travel of the valves is quick and full in opening or closing. As the valve rod moves on a straight line, and the arm T on an arc of a circle, it is provided with a sliding steel block which compensates and corrects this movement.

The exhaust chambers are provided with false seats, and the steam valve seats being part of the steam chest they are readily accessible.

The gears, cams, eccentrics and cut-off arms are all secured with set screws, and are therefore easily adjusted.

In addition to the gauge employed for verifying the length of rods and locating the steam valves, there are scribe marks 1 and 2 on bridle G (Figs. 380 and 383.) Mark 1 should appear just at the top edge of guide W for correct length of valve rod, and at Q when the port is just opening, and also indicates the amount of lap the valve has if it properly covers the seat.

TO SET THE VALVES.

In setting the valves see that the miter wheels H H are in their proper positions and secured, so as to turn the side shaft A in the direction indicated by the arrows. Then put the engine on the dead center toward the cylinder or head end, and then see that the clutch V is in engagement, making the two sections of side shaft A practically one shaft and secure the clutch. Now turn the steam eccentric at the head end, in the direction the side shaft should run, until the lower mark on the bridle G (Fig. 383) has come $\frac{1}{16}$ of an inch above the upper edge of the guide W, then secure this eccentric firmly to the shaft by tightening its set screw. With the engine in the same position, adjust the exhaust valve at the opposite or crank end of the cylinder. The crank should be turned in the same direction as that in which the steam eccentric was turned, until the full length of the exhaust gauge *e g* (Fig. 382), and the distance from the exhaust chamber to the center of the wrist Z coincide, thus causing the exhaust valve to be at the commencement of its opening movement. Next, put the engine on the crank end dead center, and set the other valves in the same manner.

TO SET THE TRIPS.

Turn the side shaft by the hand wheel H (Figs. 378 and 379) until the steam eccentric is at its full throw, or full port opening of the valve, and secure, by the set screws, the arm I, with its tripping disc,

to the cut-off shaft J, so that it will trip the valve at that point. If the steam valves do not close at the same piston travel on each end, the difference can be corrected by adjusting the discs on the arm I (Fig. 379.) When the engine is running, notice that when the steam valve is closed, that the upper mark on the bridle G is just visible at the top of the guide W. If it is not visible, it can be made to show above or disappear below the guide, by turning the air cocks in the dash pot. If this does not correct the trouble, see if there is anything under the piston of the dash pot to prevent seating. This concludes the valve setting. Of course, for absolute correct and nice adjustment, the use of the indicator will be required.

The speed of the engine can be varied by adding to or taking weight from the governor balls, or moving the sliding weight on the arm Y in or out (Fig. 379); adding weight to governor balls, or moving weight on arm Y out, increases the speed of the engine, while taking weight from the governor balls, or moving the weight on arm Y in, decreases the speed of the engine.

THE CROSS HEAD.

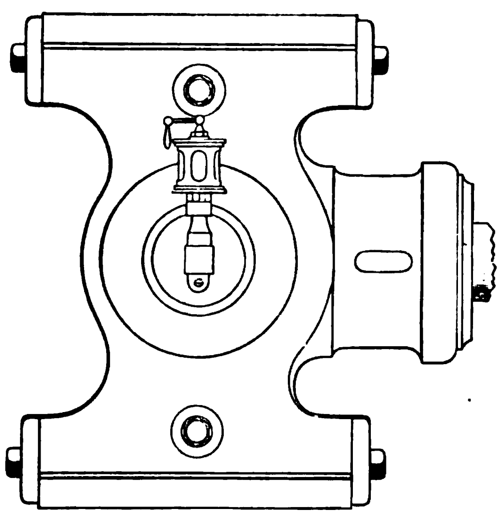


Fig. 385.

Fig. 385 is a side view and Fig. 386 is an end view partly in section. It will be observed that the wrist pin of the cross head, as shown in Fig. 385, is exactly in the center of the bearings, and hence the angular thrust of the connecting rod produces a uniform pressure on the entire surface of the bearings. The shoes of the cross head are adjustable, as shown in Fig. 386, by means of wedges.

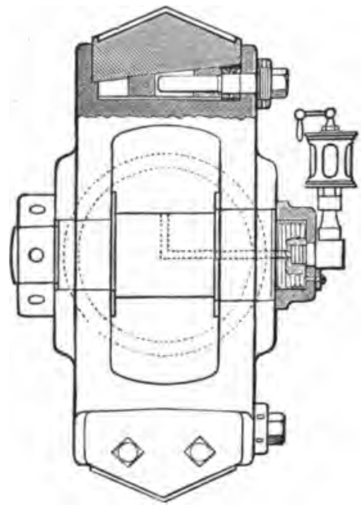


Fig. 386.

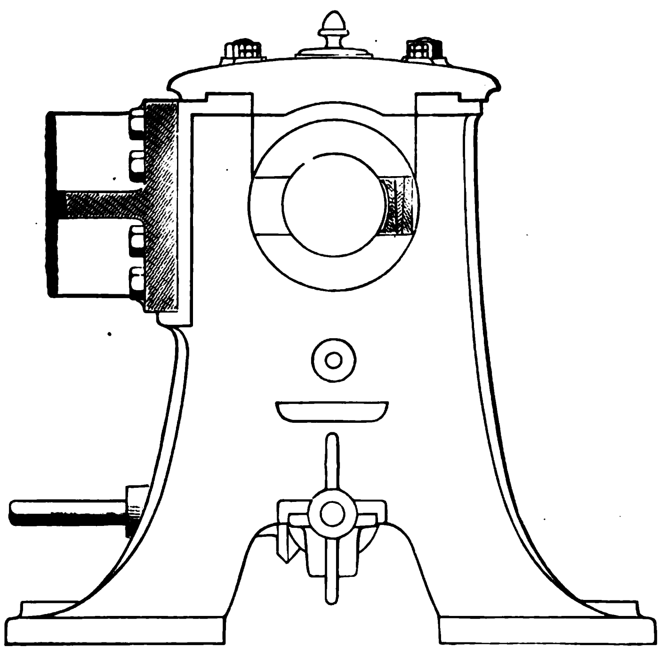


Fig. 387.

THE MAIN BEARING.

Fig. 387 represents the main bearing. It is provided with quarter boxes, which are adjustable by wedges and draw screws for taking up lost motion. This bearing also has a large base, similar to those under the slides and cylinder.

Fig. 388 is a plan and elevation of this engine.

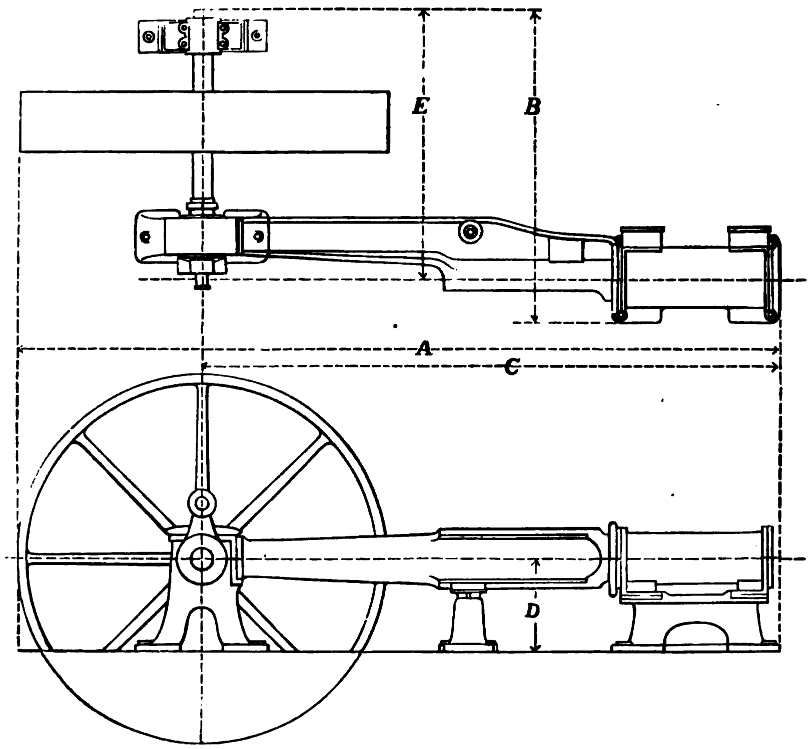


Fig. 388.

PERFORMANCE OF THE ENGINE.

To show the performance of this engine in every day practice, the following indicator diagrams are given:

The frictional diagram (Fig. 389) was taken with a Thompson indicator from a Greenwald Brown engine, at the power station of the Mount Auburn Electric Railway, Cincinnati, Ohio.

Cylinder, 42 inch stroke and 18 inches in diameter.

Boiler pressure, 60 pounds.

Revolutions per minute, 70.

Spring, 30.

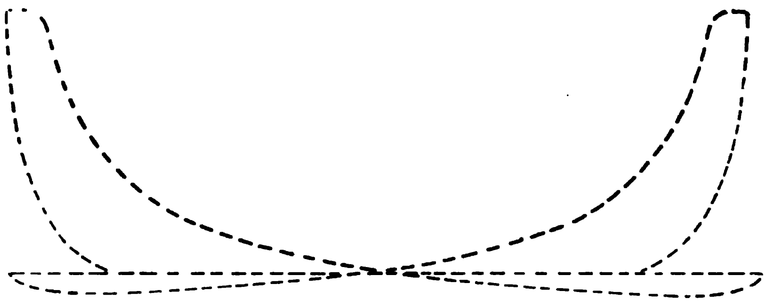


Fig. 389.

The diagram (Fig. 390) was taken with a Thompson indicator, from a Greenwald Brown engine, at The Windisch-Muhlhauser Brewing Co.'s Brewery, Cincinnati, Ohio.

Cylinder, 42 inch stroke and 20 inches in diameter.

Boiler pressure, 80 pounds.

Revolutions per minute, 60.

Spring, 40.

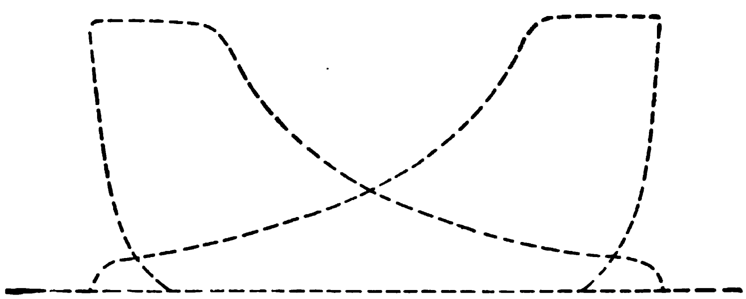


Fig. 390.

STEAM AND EXHAUST PORTS.

AREA OF STEAM PORTS.

RULE.—Multiply the area of cross section of bore of cylinder by the number of feet the piston travels per minute, and divide the product by the constant 6000, and the quotient will give the required area of steam port.

Example.—Let 12 inches equal diameter of bore of cylinder.

Let .7854 equal a constant.

Let 90 equal number of revolutions of crank per minute.

Let 3 feet equal length of stroke of piston.

Let 2 equal number of strokes of piston for each revolution of crank.

Let 6000 equal a constant.

Then we have:

$$\frac{12^2 \times .7854 \times 90 \times 3 \times 2}{6000} = 10.178784 \text{ square inches.}$$ Required area of steam port.

Performing the operation, we have:

12 inches. Diameter of bore of cylinder.
12 inches. Diameter of bore of cylinder.

24
12

"Product No. 1." 144 Square of diameter of bore of cylinder.
.7854

3 1416
31 416
78 54

"Product No. 2." 113.0976 square inches. Area of cross section of bore of cylinder.
90 Number of revolutions of crank per minute.

"Product No. 3." 10178.7840
3 feet. Length of stroke.

"Product No. 4." 30536.3520
2 Number of strokes of piston for each revolution of crank.

6000) 61072.7040 (10.178784 square inches. Required area of steam port.
6000

10727
6000

47270
42000

52704
48000

47040
42000

50400
48000

24000
24000

AREA OF STEAM PORTS—RULE IN DETAIL.

RULE.—First, square the diameter of the cylinder in inches, and call the product "Product No. 1."

Second, multiply "Product No. 1" by the constant .7854, and call the product "Product No. 2."

Third, multiply "Product No. 2" by the number of revolutions of the crank per minute, and call the product "Product No. 3."

Fourth, multiply "Product No. 3" by the length of the stroke of the piston in feet, and call the product "Product No. 4."

Fifth, multiply "Product No. 4" by the number of strokes (2) of the piston for one revolution of the crank, and call the product "Product No. 5."

Sixth, divide the last product ("Product No. 5") by the constant 6000, and the quotient will give the area or required number of square inches for steam port.

AREA OF EXHAUST PORTS FOR ENGINES HAVING EXHAUST VALVES.

RULE.—Perform the operation precisely in the manner prescribed for steam ports, until the last product is reached, then, instead of using 6000 as a divisor, use the constant 4000 as a divisor, and the quotient will give the required area in square inches for exhaust ports.

Taking the last product in the preceding example, and we have:

```
4000) 61072.7040 (15.268176 square inches. Required area of
      4000                                  each exhaust
      ----                                  port.
      21072
      20000
      -----
       10727
        8000
       -----
        27270
        24000
        -----
         32704
         32000
         -----
          7040
          4000
          -----
          30400
          28000
          -----
           24000
           24000
           -----
```

AREA OF EXHAUST PORTS—SIMPLE RULE.

RULE.—Multiply the area of steam ports, as determined by the preceding rule, by 6, and divide the product by 4, and the quotient will give the required area of exhaust ports.

Example.—Let 10.178784 square inches equal required area of exhaust port, as determined by the preceding rule.
Let 6 equal a constant.
Let 4 equal a constant.

Then we have:

$$\begin{array}{r} 10.178784 \\ 6 \\ \hline 4)\ 61.072704 \\ \hline 15.268176 \end{array}$$ square inches. **Required area of exhaust ports.**

DIMENSIONS OF STEAM AND EXHAUST PORTS.

RULE.—Assume any convenient length, in inches, for steam port, according to the diameter of the cylinder, and divide the area as determined by the preceding rules by the assumed length of port, and the quotient will give the required width of steam or exhaust port.

Example.—Let 10.178784 inches equal required area of steam ports.
Let 10 inches equal assumed length of steam ports.

Then we have:

$$\frac{10.178784}{10} = 1.0178784 \text{ inches.}$$ **Required width of steam port.**

Performing the operation in the ordinary way, we have:

$$\begin{array}{l} 10)\ 10.178784\ (1.0178784 \text{ inches.} \\ 10 \\ \hline 17 \\ 10 \\ \hline 78 \\ 70 \\ \hline 87 \\ 80 \\ \hline 78 \\ 70 \\ \hline 84 \\ 80 \\ \hline 40 \\ 40 \\ \hline \end{array}$$ **Required width of steam port.**

We now have a steam port: Width, 1.0178784 inches; length. 10 inches.

The same rule applies to exhaust ports.

Example.—Let 15.268176 inches equal area of exhaust ports.
Let 10 inches equal assumed length of exhaust ports.

Then we have:

```
10) 15.268176 (1.5268176 inches.  Required width of exhaust ports.
    10
    --
     52
     50
     --
      26
      20
      --
       68
       60
       --
        81
        80
        --
         17
         10
         --
          76
          70
          --
           60
           60
           --
```

ENGINES HAVING SEVERAL STEAM AND EXHAUST PORTS.

The Brown engine, it will be observed, has three steam and three exhaust ports for each end of the cylinder, in such a case each port should have an area of but one-third of that required for engines having but one steam and one exhaust port for each end of the cylinder, as, for instance, the Corliss engine. In the case of engines like the Brown engine, for example, where three steam and three exhaust ports are employed for each end of the cylinder, or, for that matter, any other number of ports more than one, the area required for one port is divided by the number of ports contained for the admission or the exhaust of steam. Taking the area of steam ports, as shown in the preceding examples, 10.178784, and dividing the same by the number contained at either end of the cylinder (3), we have:

```
3) 10.178784
   ---------
   3.392928 square inches.  Area required for each steam port.
```

WIDTH OF STEAM PORTS.

Now, as the ports of the Brown engine are necessarily shorter than in cases where they extend across the cylinder, we will assume the required length to be 7 inches, then we divide the area by the assumed required length, and we have:

```
7) 3.392928
   --------
   .484704  Decimals of an inch. Width of each steam port.
```

WIDTH OF EXHAUST PORTS.

After having determined the width of steam ports, and the exhaust ports are of the same length, divide the width of the steam ports by 4 and multiply the quotient by 6.

Example.—Let .484704 decimals of an inch equal width of steam port.
Let 4 equal the divisor.
Let 6 equal the multiplier

Then we have:

$$\left(\frac{.484704}{4}\right)\times 6 = .727056. \text{ Decimals of an inch. Required width of exhaust ports.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{r} 4)\ .484704 \\ \hline .121176 \\ 6 \\ \hline .727056 \end{array}$$ Decimals of an inch. Required width of exhaust ports.

CHAPTER XXII.

THE BUCKEYE ENGINE.

This engine presents many novel and interesting features worthy of consideration by the student of steam engineering. Many of these features are peculiarly its own, and tend to place it in a class separate and distinct from other leading American steam engines. As this engine has come into general use it is a matter of importance to engineers to understand its construction and the adjustment of the valve-gear, as well as the running gear generally. It is a high speed engine and automatic in its valve movements. In order that the student may have a clear conception of the anatomy of this engine, a detail description of its various parts, with illustrations, are here given.

PUTTING THE ENGINE TOGETHER.

SHOP MARKS.

All parts capable of being wrongly put together or misadjusted, are adjusted in the shop as correctly as possible and marked, so that in case any adjustment has been disturbed, by shipment or otherwise, it can readily be restored.

THE MAIN ECCENTRIC.

The main eccentric is marked to indicate its angular position on the shaft by a chisel mark on its hub next to the governor eccentric, and a corresponding mark on the shaft. But these marks being invisible when the governor eccentric is in position, the engineer should make marks on the eccentric and its strap to coincide when the crank is placed on its outer dead center. The outer dead center is preferred to the other because in that position the mark on the eccentric may be located near its longest radius and be in convenient position for observation. When set to the marks the leads, as shown by marks, will be from $\frac{1}{16}$ inch to $\frac{5}{16}$ inch, and compressions from $1\frac{7}{8}$ inches to $5\frac{1}{4}$ inches, according to size of engine, generally about one-eighth stroke.

THE GOVERNOR CONTAINING WHEEL.

The governor containing wheel is similarly marked on its back hub, and when this mark coincides with a corresponding mark on the

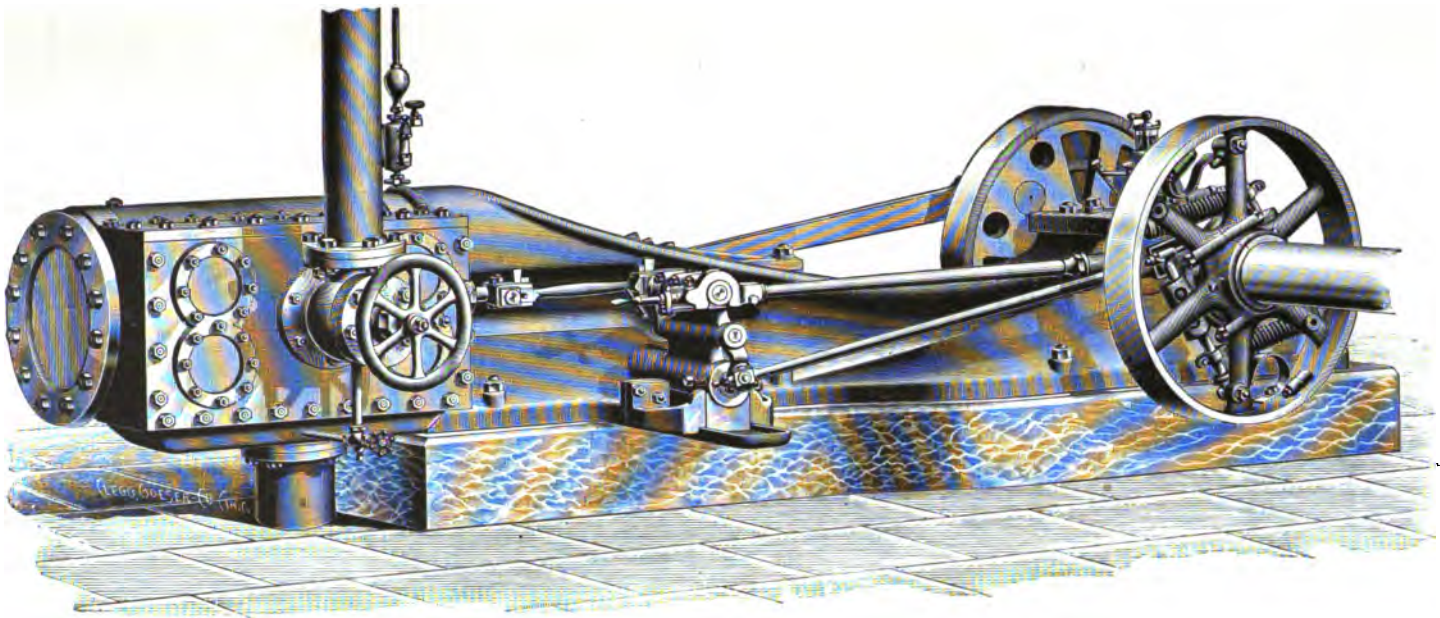


Fig. 391.

BUCKEYE ENGINE—BACK VIEW.

shaft, and other adjustments are correct, the latest cut off will generally be at about five-eighths of the stroke, and the earliest somewhere inside of the first inch or two of stroke, which, when non-condensing and under ordinary conditions as to pressure, friction, etc., will hold the engine from running objectionably above speed under its lightest possible load and highest pressure. This is the safety test, the failure of which is corrected by slightly advancing the governor containing wheel on the shaft. On the other hand, the latest cut off will give maximum guaranteed power, with margin for regulation.

To obtain greater maximum power, the governor containing wheel may sometimes be set a little back of the marks, as for instance, when disconnection from a considerable portion of load is impossible, or when the pressure carried is lower than usual, but in all cases of such adjustment the safety test should be applied.

THE ECCENTRIC RODS.

The eccentric rods are marked to indicate the distance they enter the bases of the eccentric straps, and are just visible when correct.

These and other marks should always be followed except when performance tests, hereinafter given, show reasons for departure from them.

ADJUSTMENTS INDEPENDENT OF MARKS.

It is very essential that every engineer should thoroughly understand the adjustment of any engine under his charge, by means of inspection of the performance of the parts concerned, independent of shop marks, as such marks may become obscured, obliterated, or tampered with, or may not indicate the best possible adjustment for existing conditions.

THE MAIN VALVE.

The main valve differs from the common slide valve in that it delivers steam to the cylinder from its interior through ports in its face, and exhausts steam from its cylinder at its ends into and through the containing chest, hence its movement is the reverse of that of the common slide valve; it admits steam to the cylinder by moving toward the end to which steam is to be admitted, and from the end from which steam is to be exhausted, and consequently, the openings for admission are not visible on removal of the chest cover, but are shown by marks on the cylinder and valve in line with the borders of their respective ports.

The marks on the valve, however, represent its ports only at the ends where narrowest, and ignore a certain amount of extra graduated lead introduced to moderate the shock of induction at high pressure: hence when lead is spoken of, that shown by the marks is meant.

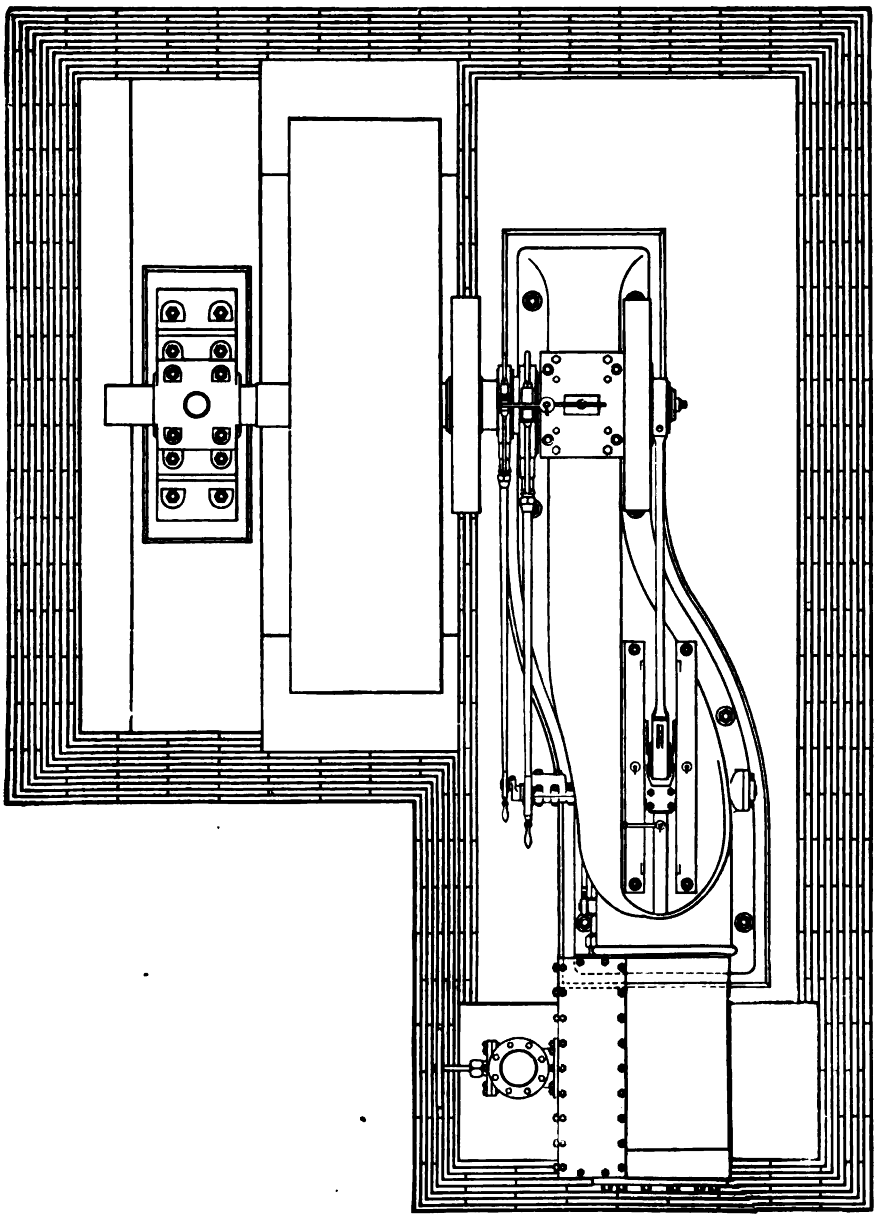


Fig. 392.

PLAN.

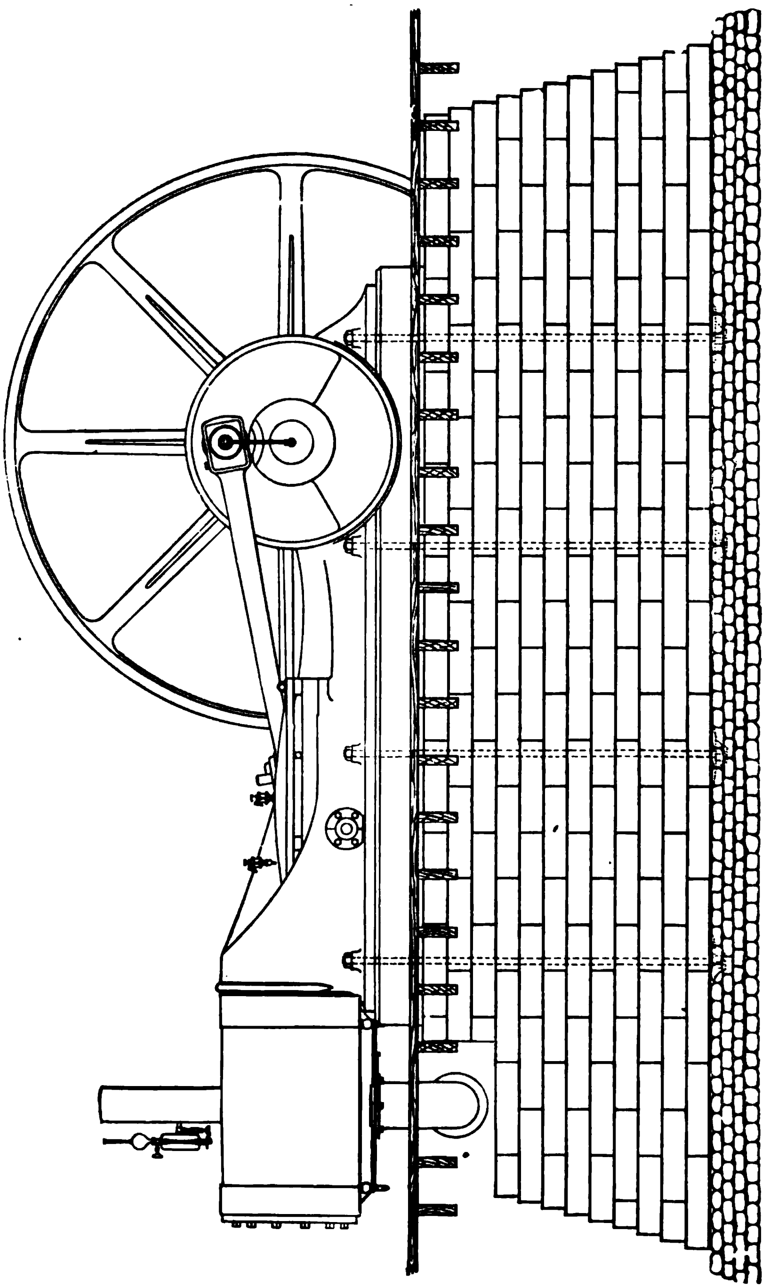


Fig. 398.
FRONT ELEVATION.

The exhaust opening, however, is visible like the admission of the common slide valve, and for which it must not be mistaken.

TO SET THE MAIN VALVE.

If the main eccentric is not correctly set the valve can be set only to give equal openings for admission at the two ends, regardless of the

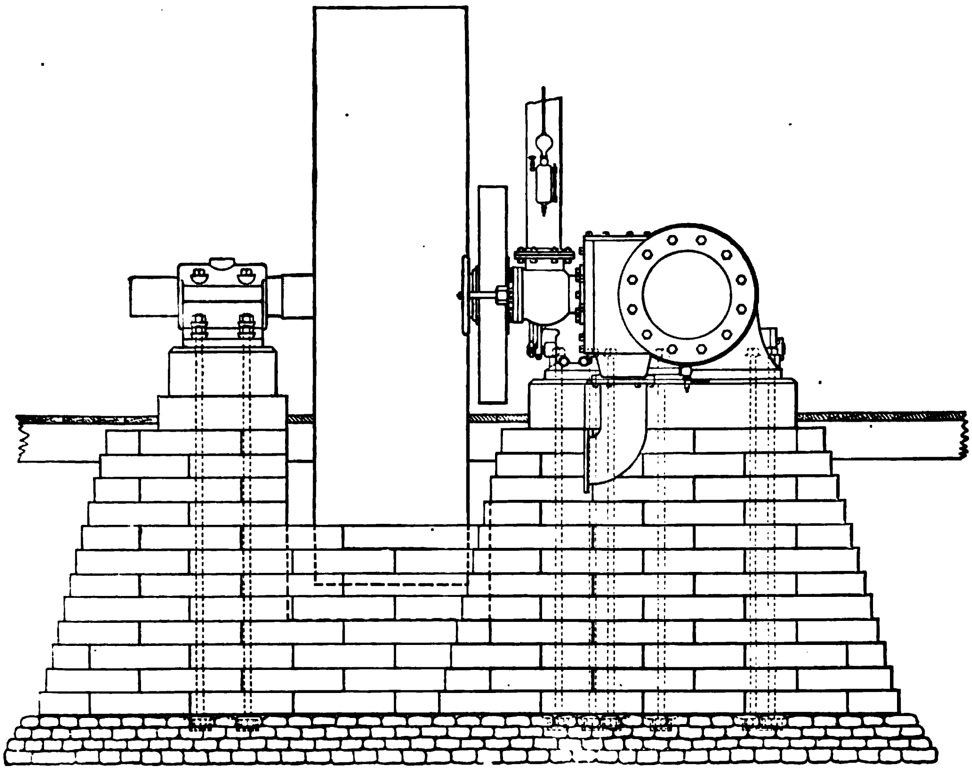


Fig. 394.
END ELEVATION.

position of the piston; these once equalized will remain so in all positions of the eccentric, though equalization should be tested after the eccentric is set by noting whether compressions are equal.

TO SET THE ECCENTRIC.

If the engine runs over—that is, top of driving wheel from the cylinder—place the crank on its outer dead center, otherwise, on its inner dead center. In either case the projecting side of the eccentric should be upward but inclined toward the crank sufficiently to cause the valve to show the desired lead; but if the engine is so conditioned,

as to size and attachments, that it can be readily revolved, the compression should next be noted, both as to equality and amount; if unequal they should be equalized even at the expense of unequal leads, since compression with high speed and high duty engines is a much more important factor in its effects in the running qualities of the engine than lead.

THE AMOUNT OF COMPRESSION.

The amount of compression may be, as before stated, about one-eighth of the stroke, though with the high speed short stroke styles, more may be frequently given to advantage.

TO ADJUST FOR AMOUNT OF COMPRESSION.

To adjust for the amount of compression, shift the eccentric forward for more or backward for less compression; but any considerable change of this kind will be accompanied by a slight loss of equality, owing to the fact that the piston travels faster when approaching the end of its travel farthest from the shaft than at the other end.

THE ADJUSTMENT FOR EQUALIZING COMPRESSIONS.

The adjustment for equalizing compressions is to either lengthen or shorten the eccentric connection by adjusting the rod out of or into the stroke, or to shift the member clamped to the valve stem, so as to throw the valve toward the end at which the compression is least. In chosing between the above adjustments select that which will cause the rocker arm to be nearest vertical at its end movement, bearing in mind, however, that if the cut off has been equalized the first named adjustment will not disturb that equality, while the last named shifting member on stem will.

TO SET THE GOVERNOR WHEEL.

Place the crank on one of its dead centers, preferably the inner one, and set the governor so that its eccentric will be on its inner dead center also, as nearly as can be determined by inspection. The shop marks will then coincide, or nearly so, unless the eccentric has been connected to the levers a half turn, relatively to the wheel, from its position when marked, in which case the marks will be diametrically opposite each other, hence, the need for observing the above rule rather than placing entire reliance on the marks.

CUT OFF EQUALIZATION.

It remains now only to equalize the cut off as between the two strokes, and it should be understood at the outset, that this can not be done perfectly for all points within the range of variation, on account

of unequal piston velocity at corresponding points in outer or inner strokes, due to the virtual shortening of the connecting rod when deflected from parallelism with the center line. But if equalized at or near quarter stroke, it will be practically equal throughout its proper working range, the inequality becoming apparent only at very early cuts, when that at the front or shaft end will be latest.

ADJUSTMENT FOR EQUALIZATION.

Block the governor wheels about half way out, turn the engine over and note when, in each stroke, the cut off takes place, measured from its beginning. If found unequal change the length of the connections between the governor eccentric and valve stem, in such a way as to throw the valve toward the end at which the latest cut off takes place. The adjustment may be made at the junction of the eccentric, straps, and rod, or in the connection between the upper cut-off rocker arm and the valve stem. If the latter place is chosen, two different constructions may be encountered. Originally the connections screwed out of or into the neck of a bronze box, fitted to a pin in the rocker arm above mentioned, and was secured by means of a lock nut. Laterally the box and rod are made in one piece, and the adjustment is made by screwing the ball and socket joint to and fro on the rod, securing with lock nut as before.

CHOICE OF METHODS OF THE ADJUSTMENT.

Choice of methods of the adjustment should be made by noting which will cause the pin in the upper cut-off rocker arm to vibrate most nearly equally each way past the pin in the main arm near which it works. Any noticeable inequality in that respect should be corrected by adjustment at the eccentric strap junction first, and cut off equalized at the other place afterward.

TO DETERMINE POINT OF CUT OFF OF VALVE CLOSURE.

If there is no steam, remove the cover plates of the balance pistons, when the closure may be seen, or if not visible up to closure, it may be felt for by means of a piece of wire at its end like a chisel, and it should be considered closed when slight traces of an opening can still be felt. If there is steam available remove the indicator plugs, admit enough steam by the throttle to blow a little at the opened orifices, but not enough to move the engine, turn the engine over slowly until the point of cut off is indicated by the stoppage of the blow, or preferably by its partial stoppage, since if entirely stopped, the point may have been passed by a small but unknown amount. This is perhaps the most exact method, especially if the engine is stopped

in each stroke, when a slight forward strain on the eccentric by its balance weight will stop the blow, and a push in the contrary direction will start it again.

Having determined the points of cut off and found them unequal, proceed as before directed; but repeated trials may be avoided by placing the cross head at the mean of the two unequal points, preferably near the latest, and adjusting as required.

When an engine is too large, or otherwise so conditioned as to preclude turning over, loosen the governor wheel and revolve it on the shaft. With the back cover plates removed note how far the valve, when at each extreme of its travel, must move to close the open port. That distance should be the greatest at the stem end by an amount equal to one-twelfth of the travel of the valve. This displacement from symmetrical movement is necessary to compensate for the distortion of piston travel before explained.

ADJUSTMENTS BY INDICATOR CARDS.

When cards for careful adjustment are taken, it is important that the movement rig be substantially correct, and that such slight distortions as may take place shall be the same at each end, so as not to introduce apparent inequality when none exist.

The rigs now, and for a number of years, furnished with these engines are correct enough for the purpose, though the following test, if carefully made, would show slight distortion. With engine on one dead center, and indicator properly coupled up, apply the pencil to the paper and make a short vertical mark, such as the spring will permit without undue force. Then turn the engine till the cross head moves one inch and make another mark, and so on throughout the stroke, ending with the other dead center. On measuring the scale so produced the end divisions will be found slightly larger than the others, but if the distortion is the same at each end no harm will result, but in order that they shall be equal it is necessary that the angle formed by the chord, with the center line of its actuating lever, shall be 90°—a right angle—at mid stroke. Any deviation from this condition will diminish the divisions at one end and shorten the diagram.

DISTORTION FROM ELASTICITY OF THE CORD.

This is to be apprehended only at high speeds—200 revolutions and upward per minute—and is present when a diagram taken at full speed is longer or differentially located on the paper than when taken at slow speed. To avoid this use a well-stretched cord, having as little elasticity as possible, or, better still, a light wire; and, in either case, give the drum springs as much tension as may be thought safe.

The indicator should be attached by means of ½ inch pipe fittings and a three-way cock, so that diagrams from each end can be taken in quick succession; and when taking cards to test cut off equilization the cock should be reversed several times, applying the pencil each time, so as to avoid being deceived by changes of load. If this method results in a multiplicity of lines, so much the more evident is the need of it, and if the diagram is not thought sufficiently reliable, continue taking until one is obtained that covers a period of sufficient uniform load to be reliable.

TEST DIAGRAMS.

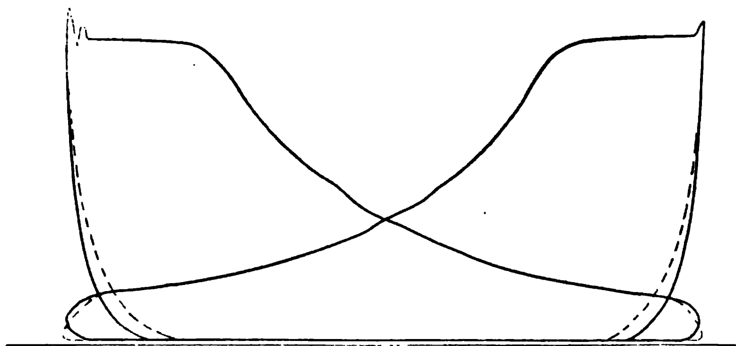


Fig. 395.

Fig. 395 represents perfect performance as to equality of the several functions, and in nearly perfect performance in other respects as is practically attainable, especially at high rotative speeds. The dotted compression and exhaust lines represent about the amount of allowable variations in these functions to meet different conditions, mainly rotative speed, as the higher the speed the greater the compression that may be needed to give best running qualities, by which test a greater amount even than that shown by the dotted lines is sometimes found to be advantageous.

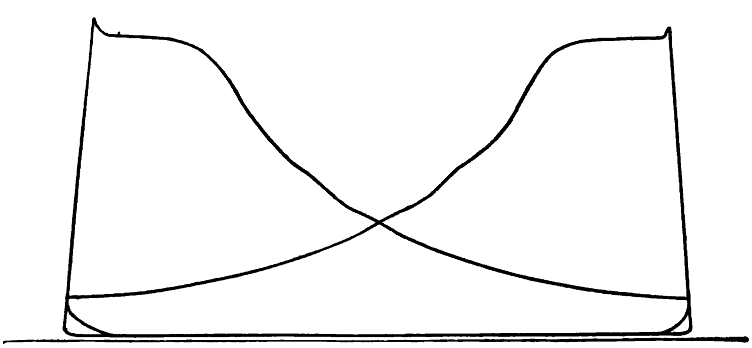


Fig. 396.

Fig. 396 shows a "slow" main eccentric, about such as might be produced if the eccentric had no angular advance; that is, if it was at its mid travel when the crank was on the dead centers. The remedy is to advance the eccentric until satisfactory performance is attained—good diagram and running qualities.

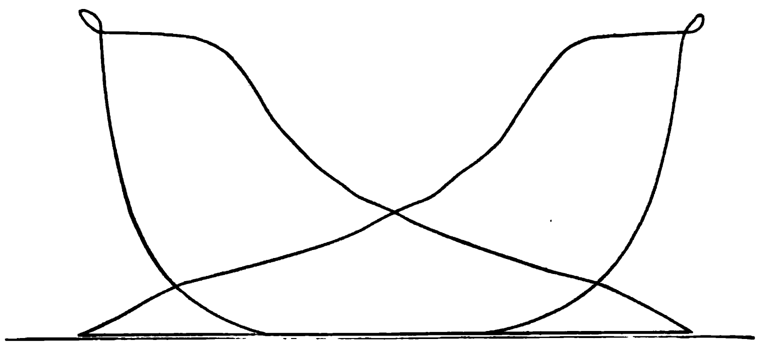


Fig. 397.

Fig. 397 shows a fast eccentric—that is too much angular advance. The remedy is to set it backwards until satisfactory diagrams are obtained, though, as a matter of fact, if the displacement from correct and equal adjustment were that of the eccentric only, the distorted result would show slight inequality.

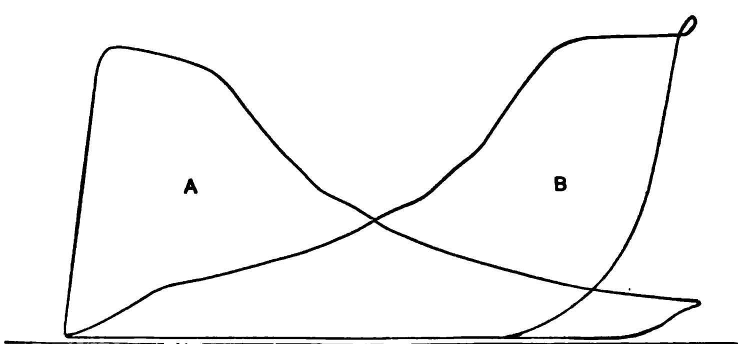


Fig. 398.

Fig. 398 shows displacement of the main valve, giving great inequality in its functions. The remedy is to make the proper adjustment for equalization, according to the method already explained, throwing the valve more toward the end from which diagram A was produced.

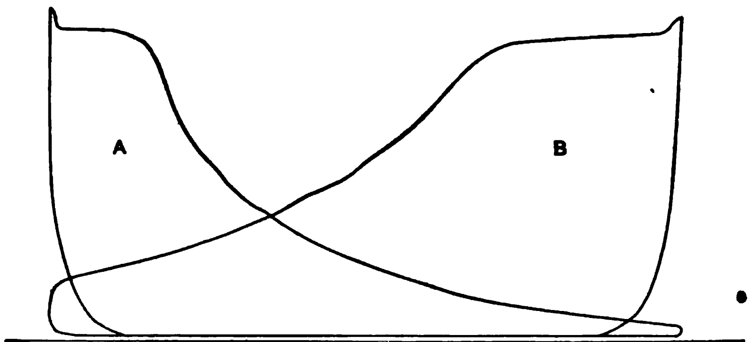


Fig. 399.

Fig. 399 shows unequal cut off. The remedy is to make adjustments as previously explained, throwing the cut-off valve toward the end from which the largest diagram B was taken.

THE GOVERNOR.

The following names of details of the governor, as shown in Fig. 400, are given for convenient reference:

The levers or weight arms *a a* are called levers for convenience.

The weights A A, clamped on the levers *a a*, determine speed, the heavier they are the slower the speed.

The lever pivots *b b* are studs screwed to arms of the containing wheel on which the levers move freely.

The links B B couple the free ends of the levers *a a* to ears on the sleeve of the governor eccentric C.

The governor eccentric C is free to turn on the shaft, and is advanced 90° by the outward movement, or expansion of the levers to the outer extreme of their range of movement.

The main springs F F are of steel wire, tempered. They are attached adjustably to the arm of the containing wheel by means of the tension screws *c c*.

The tension screws *c c* are the means by which the tension of the springs is adjusted.

The spring clips *d d* are clamped on levers *a a*, and are provided with slots or eyes into which the springs F F are hooked, and they are adjustable along the levers to a certain extent.

The outer lever stops *e e* are cylinders of wood fitted to sockets in the outer caps of the links B B, and they limit the expansion of the levers by coming in contact with the inner surface of the wheel rim; but when the governor works properly they seldom if ever strike.

The inner lever stops *f f* are blocks of wood on which the levers rest when not expanded.

The auxiliary springs P P are employed to counteract a tendency to "race" when the tension of the main springs is great enough to give close regulation at light and medium loads. They should be of just sufficient force to start the levers out at very near the running speed,

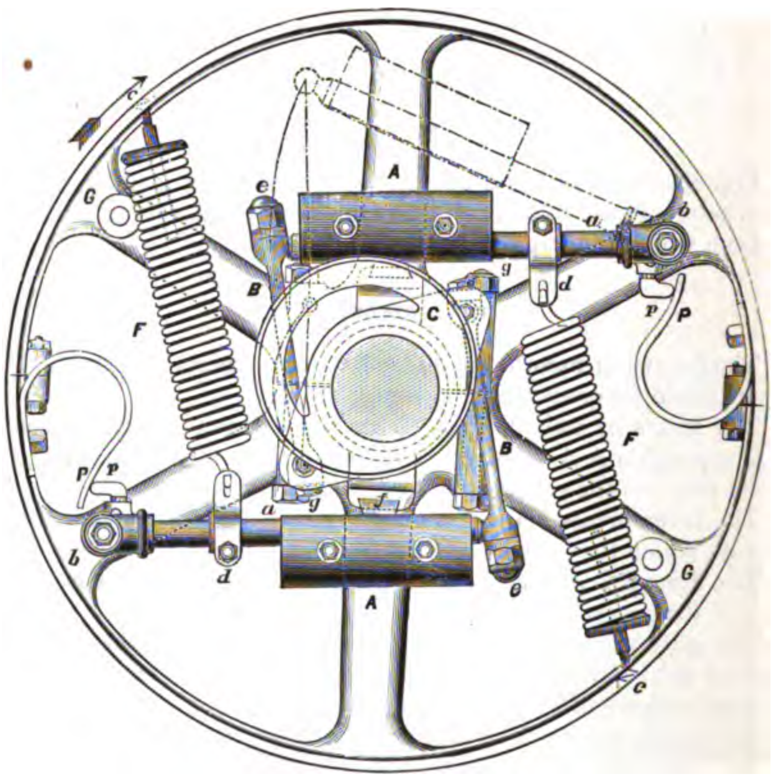


Fig. 400.

for without them the levers would not start out until running speed was exceeded. With sufficiently strong main springs the auxiliary springs are not needed, but the strains and pivot pressure resulting from such spring strength are objectionable. When used the tension of the main springs should always be so high as to require their use to prevent "racing," otherwise speed will be too slow at heavy loads. They have contact with the fingers *p p* when the levers are about half way out.

The guide rollers G G are introduced in most high speed engines to restrain the springs from bowing outward from centrifugal force,

and so changing the direction of pull on the clips as to disturb regulation; but they are seldom needed at speed below 240 revolutions per minute.

CLOSE REGULATION.

Ignoring all frictional or other disturbing elements, perfect regulation would be secured by such an amount of initial tension on the main springs F F that if the arms could move inward until their center of force coincided with the center of the shaft, or a line joining the center of the shaft and the pivots of the levers, in other words, the zero of centrifugal force, the springs would at the same time have their tension just relaxed; that is, arrive at their zero of force. Then it is evident that as the levers moved outward at a constant speed, centrifugal force and spring force, would increase in the same ratio and be in equilibrium with each other at the same speed at all points within the range of the movement of the levers. In other words, the regulation would be perfectly isochronous. But suppose the spring tension to be less than this, so that as the levers moved inward, the zero of spring force would be reached before that of centrifugal force; then as they moved outward, the spring force would increase more rapidly than the centrifugal force, at a constant speed, and a constantly increasing speed would be required to keep the two forces in equilibrium, and the amount the speed would have to increase in order to carry the levers from their inner to their outer limit of movement would be the measure of the governor's variation. Thus, if 100 revolutions per minute be required in a given time to start the lever outward, and 105 in the same time to expand them to their outer limit, the extreme variation would be 5 per cent., which would be tolerably close regulation, since in practice the changes of load seldom cover more than half that range.

TO SECURE CLOSE REGULATION.

It follows that when too much change of speed accompanies changes of load or of pressure, the remedy is to increase the spring tension until the regulation is satisfactory. This operation is accompanied by increase or outward shifting of nuts or inward shifting of spring clips to prevent increase of speed. If the equilibrium becomes unstable and "racing" makes its appearance, as shown by the levers periodically moving outward and inward without load fluctuations sufficient to produce such movements, the tension must be relaxed until equilibrium is restored; but if "racing" makes its appearance only when heavily loaded, it may be cured by giving the auxiliary springs more power by means of greater reach of fingers, or by opening them slightly, but not so as to continue in contact with the fingers much beyond mid movement.

FRICTIONAL DISTURBANCE—CENTRIPETAL FRICTION.

The friction of the straps of the governor eccentric, and of the valve and gear operated by it, is centripetal in its tendency, that is, tending toward the center. It tends to hold the eccentric backward and to pull the levers inward centripetally, like the action of the main springs, thus accelerating the speed slightly. The stronger the main springs the less this acceleration, but it is always present to some extent, and it varies throughout the range of movement of the levers, being greatest when they are at or nearest their inner stops, when the forces in equilibrium are weakest and the angle of the links and eccentric ears most acute, and next greatest when levers are most expanded, when link and ear angle is most obtuse but forces in equilibrium greatest; and least at or near mid movement, when the link and ear angle is a right angle or a little more.

But for this inequality of frictional effect it would call for only slightly less spring tension, but as it diminishes from inner extreme to mid movement, or slightly beyond, and increases from that point outward, it would be counteracted by diminishing tension for the inner half of movement and increased tension for outer half, so that if tension is sufficient to give close regulation throughout outer half of lever movement it will be slightly in excess for inner half, and the result will be that on starting the engine the levers will not start out until proper speed is exceeded, and stable regulation is possible only with loads too light to require at any time later quarter-stroke cut off, hence the functions and needs of the auxiliary springs.

TO CHANGE SPEED.

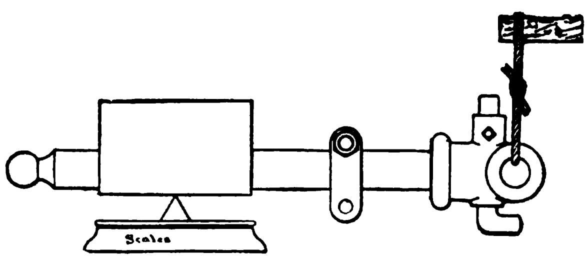


Fig. 401.

For any considerable change of speed different weights should be attached, the weight of which may be found as follows:

Remove one lever with its weight attached, suspend or support the pivot end at the pivot hole, and allow the weighted end to rest on a fulcrum on scales, as shown in Fig. 401, at a measured distance from one end.

The point selected is immaterial, but when once selected the same point must rest on the scales during subsequent weighings. A point which comes about central over the inner stop block may be preferred as coinciding more nearly with the center of gravity of the attached weights. Weigh the weighted lever thus independently supported at its pivot, multiply the weight by the square of the present number of revolutions per minute, and divide the product by the square of the desired number of revolutions per minute; the quotient will give the required weight of weighted lever, weighed as shown in the diagram

MINOR CHANGES OF SPEED INCREASE.

Increase of spring tension may be tried, if room for it exists on the tension screws, and if, when the desired increase has been obtained, there is shown no tendency to "race" at any load, and at starting the levers start out before proper speed is exceeded, the correct adjustment has been made, and the regulation will be closer than before. But if the necessary tension can not be applied, proceed as follows:

Add a little spring tension and also shift the spring clips from the lever pivots about twice as much as the added tension. The stability and closure of regulation will be about as before, for while increase of tension tends toward instability of equilibrium, shifting the clips from the lever pivots, as above described, tends toward increased stability and more speed variation.

Another method is to shift the weights toward the pivots of the levers, if the space in that direction is not thereby too nearly exhausted and the weights still have good bearings on their inner stops.

TO DECREASE SPEED.

Spring tension may be reduced, but to retain as close regulation as before, the spring clips should at the same time be shifted toward the lever pivots twice as much as the tension is reduced. Decreased speed may also be accomplished by shifting weights from pivots if room exist for such movement

REVERSAL OF DIRECTION OF MOTION.

Place lever pivots in the idle holes, or holes occupied by guide rollers, with levers turned end for end and other side out, so that after motion is reversed the pivot ends of levers will still go ahead of the free ends, and the weights will rest on the stops as before. Reset containing wheel and eccentric according to directions already given.

VALVE ARRANGEMENT.

Fig. 402 shows a central section through the cylinder and the right-hand end of the valve and chest, the other being a section through one of the balance pistons. The line of section for valve and chest is indicated in Fig. 403 by dotted lines *b b f f b′ b′*.

The steam enters at D (Fig. 402) into passage *a a*, which lead to balance pistons *d*, through which it passes to the interior I I of the valve, as shown by the arrows, in which chambers the boiler pressure is constantly maintained when the engine is at work. The balance pistons are packed in their orifices, with steam metal spring rings and followers F, and also fitted to work steam tight on faces on the cover plates of the valve, thus forming a steam tight communication between passages *a a* and the moving valve. Coiled steel springs E serve to hold the pistons to their seats on the valve when the steam is shut off.

From the interior of the valve the steam is admitted to the cylinder through ports in its faces, as they are alternately brought by its movement to coincide with the cylinder ports. The valve is shown as just beginning to admit steam at the left-hand end, as indicated by the arrows; while at the other end the cylinder port is shown partly open for the exhaust, which passes into the chest and thence into the pipe K at its bottom, its course also being indicated by arrows.

The cut-off valve is formed by two plates, shown at *c c* (Fig. 402) and *v v* (Fig. 404), and rigidly connected by rods *h h h′ h′* (Fig. 404). These plates work on seats surrounding the valve ports, which valve ports they alternately cover at times relative to the piston travel determined by the governor.

The area of the balance piston is such as to hold the valve seat against the force tending to throw it off, due to the pressure in the valve and cylinder ports. This tendency is variable, being greatest at the moment of induction and least after cut off, while the counteracting force due to the area of the balance pistons is constant, and consequently in excess, except during induction. To counteract such excess, shallow recesses corresponding to the cylinder ports in shape and area are formed in the valve seats near their inner margins. These recesses, " or relief chambers," as they are called, one filled with steam pressure from the interior of the valve through small holes, as shown at *f* (Fig, 402), at the right-hand end, while the port at the same end is open for exhaust, and relieved of such pressure by the movement of the valve, as shown at the left a little before induction. By this device the average pressure of contact is reduced to about what is needed to insure wear enough to keep the surface in good condition.

Channels *e e* (Fig. 402) are cut across the valve faces near their ports, to prevent the steam in the ports from acting on any larger area —through any slight imperfections of fit—than is embraced in the bal-

ance pistons, and thus throw the valve from its seat. To perform this function they require to be of such capacity, and the free escape of steam that may leak into them to be so well provided for, that no

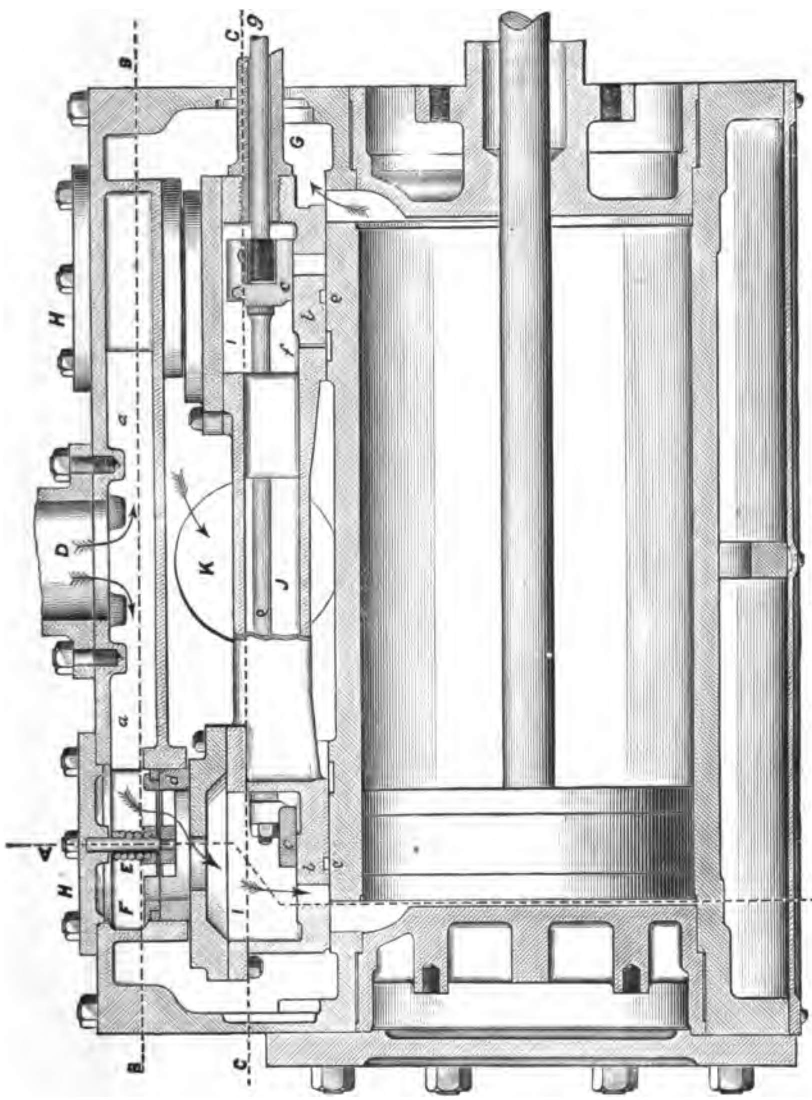


Fig. 402.

pressure in excess of that in the chest can be found in them. These conditions are attended to in the construction of these engines, but since wear from the valve forces reduces their depth, cases may occur of their becoming inadequate from long wear, so as to require deepening or otherwise enlarging.

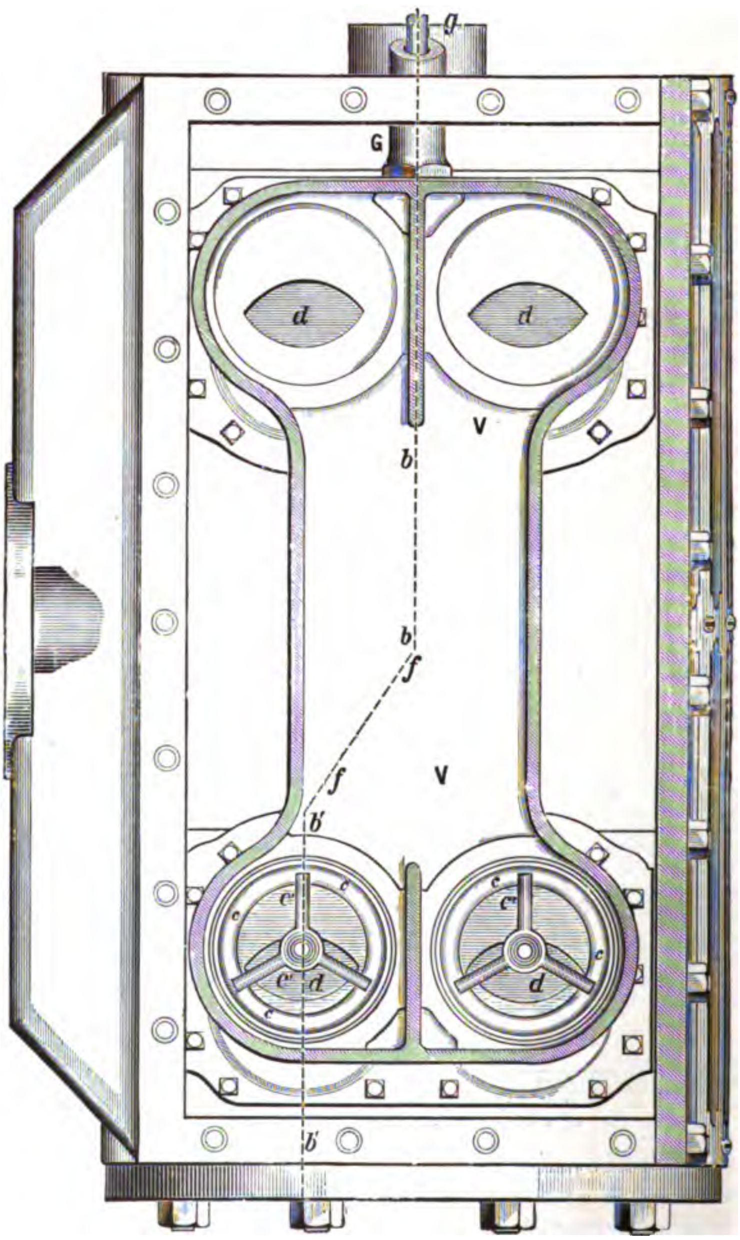


Fig. 403.
SECTION ON LINE B B (FIG. 402).

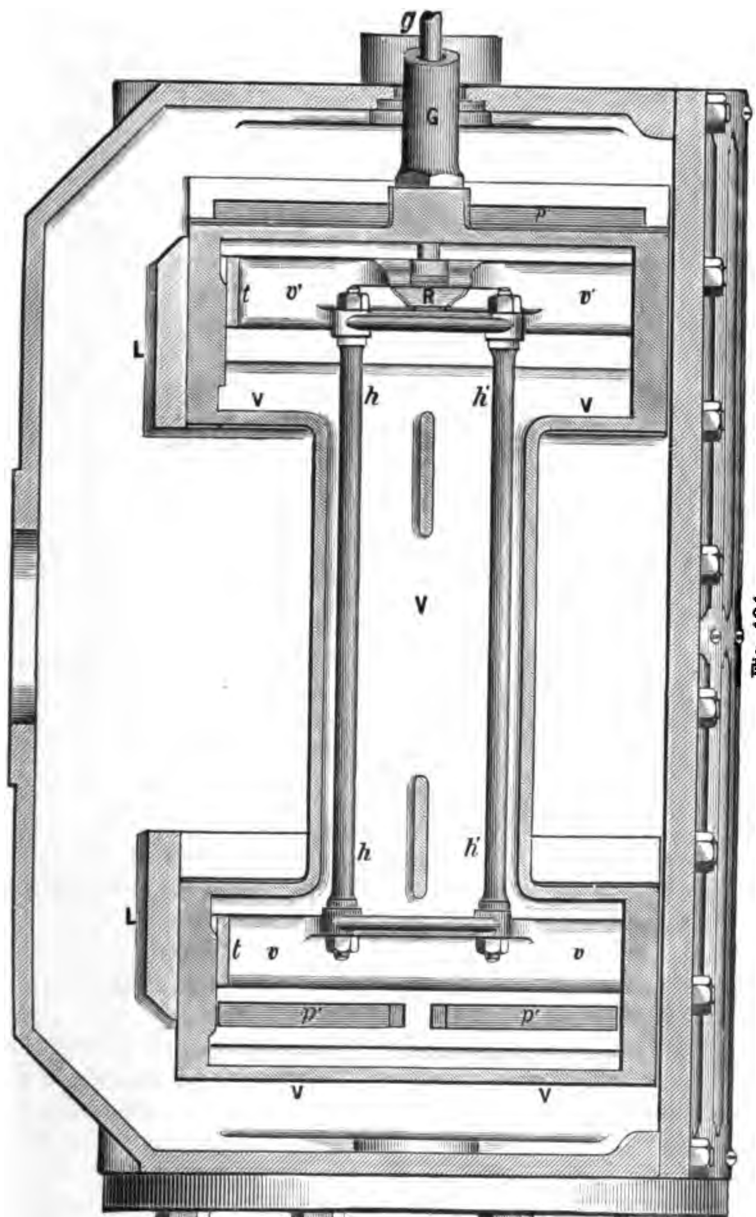


Fig. 404.
SECTION ON LINE C C (FIG. 402).

Fig. 405 represents a cross section of the girder bed and compound rocker arm. The rocker shaft *a* vibrates in adjustable bearings *b b;* the main rocker arm A is clamped to this shaft by bolts *c c*, and transmits motion from the eccentric rod to the valve stem. Through the center of the arm A there is a long bearing, carrying the cut-off rocker shaft B, with its arms C and D. This shaft receives motion from the

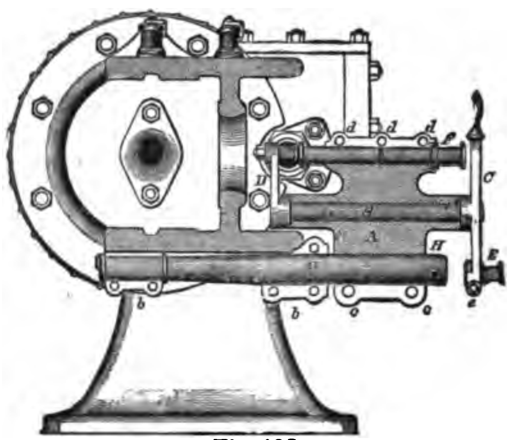


Fig. 405.

cut-off eccentric rod on the pin E; the cut-off valve thus derives a motion compounded from the movements of both eccentrics, and therefore has a constant length of travel on, and relative to, the main valve.

RUDIMENTS AND ANALYSES OF INDICATOR DIAGRAMS.

For convenient reference the following names are given to the different parts of a diagram, as shown in Figs. 406, 407, 408 and 409:

THE STEAM LINE.

The steam line, or initial pressure line, A B (Fig. 406), is formed while steam is being admitted to the piston.

In diagrams from automatic cut-off engines, shown in Fig. 406, the steam line should theoretically be horizontal, and should show a pressure nearly or quite equal to that in the boiler, but in practice, the port and steam pipe areas and valve travel necessary to attain these conditions at the high piston and rotative speeds required for electric generation, and some other purposes, would involve greater loss from friction, condensation, leakages, etc., than would result from several pounds' loss of average pressure during admission.

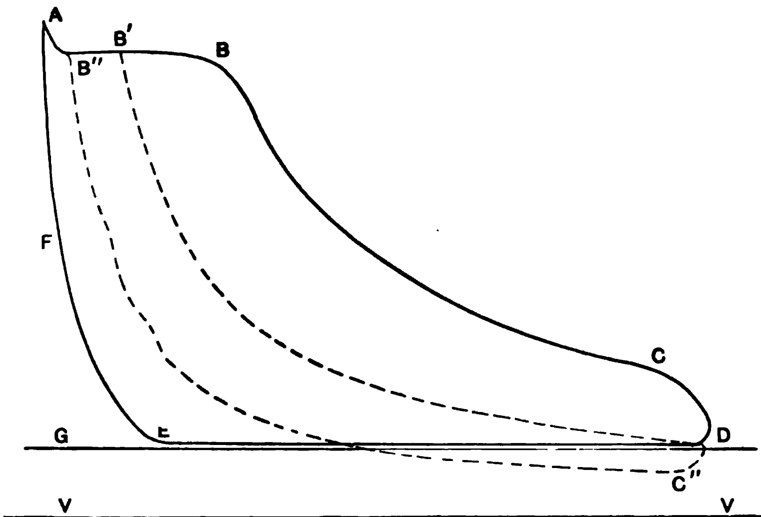


Fig. 406.

DIAGRAMS FROM THROTTLING ENGINES.

In diagrams from throttling engines, on the other hand, as shown in Fig. 407, the greater the fall of pressure during admission, the better the economy indicated, since other things being equal, this fall of pressure, which may be called initial expansion, leads to a correspondingly diminished terminal pressure.

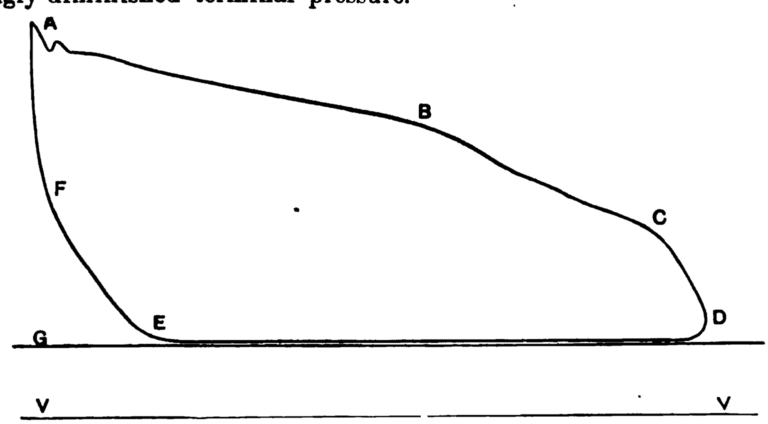


Fig. 407.

THE POINT OF CUT OFF.

The point of cut off B, being anticipated by more or less fall of pressure, it is customary to locate it at the point of contrary flexure; that is, the point where the line ceases to be convex and begins to be concave.

THE EXPANSION CURVE.

The expansion curve B C (Figs. 406 and 407) begins at the point of cut off B and terminates at the point of exhaust C. The latter, like the former, is presumably at the point of contrary flexure, except when the load is so light that the return pressure falls below the atmosphere. Such a case is represented by the dotted expansion curve B″ C″ (Fig. 406), which is about what would be produced if all load were thrown off; in other words, it would produce a friction diagram.

THE EXHAUST LINE.

The exhaust line C D represents the change of pressure which results from the opening of the exhaust. This change may, in automatic cut-off engines, be a fall or rise, or no change at all, when terminal pressure is the same as back pressure, as shown by curve B′ D (Fig. 406), in which case the exhaust line is absent.

THE BACK PRESSURE LINE

The back pressure line D E represents the pressure acting against the piston during the return stroke. With non-condensing engines it should coincide with atmospheric pressure line, or not materially exceed it—exact coincidence is seldom attained—and with condensing engines it should not materially exceed the pressure shown by the vacuum gauge. With the high pressure cylinders of compound engines it should show sufficient pressure above atmosphere to permit the next cylinder to do its proper share of the work.

THE POINT OF EXHAUST CLOSURE.

The point of exhaust closure E, being anticipated by some rise of pressure, and there being no change in the direction of curvature, can not certainly be located, but it can be approximated closely enough for all practical purposes, when a clear understanding is had of the laws governing the formation of expansion and compression curves.

THE COMPRESSION CURVE.

The compression curve E F indicates the rise of pressure due to the compression of the steam remaining in the cylinder after the closure of the exhaust. It begins at the point of the exhaust closure and ends where steam begins to be admitted, a point far more difficult to locate in such diagrams, as shown in Figs. 406 and 408, than any of the other points named, as the engines from which they were taken—Buckeye—are provided with a graduated lead, so adapted to existing conditions as to cooperate with compression in reducing to the utmost

possible degree the tendency to pound when passing the dead centers; and it is found that when such running qualities are best attained the terminal point of the compression curve is most obscure.

THE LEAD OR ADMISSION LINE.

The lead or admission line F A, when one exists, completes the diagram. Its beginning F, as previously explained, is mostly, and should be, very uncertain, but its termination at the beginning of the steam or initial pressure line is the most clearly defined point of all, except as it may seem obscured by vibrations of the pencil lever of the indicator, which are always present at high speeds and with a free working instrument. The line is absent when compression begins so early that

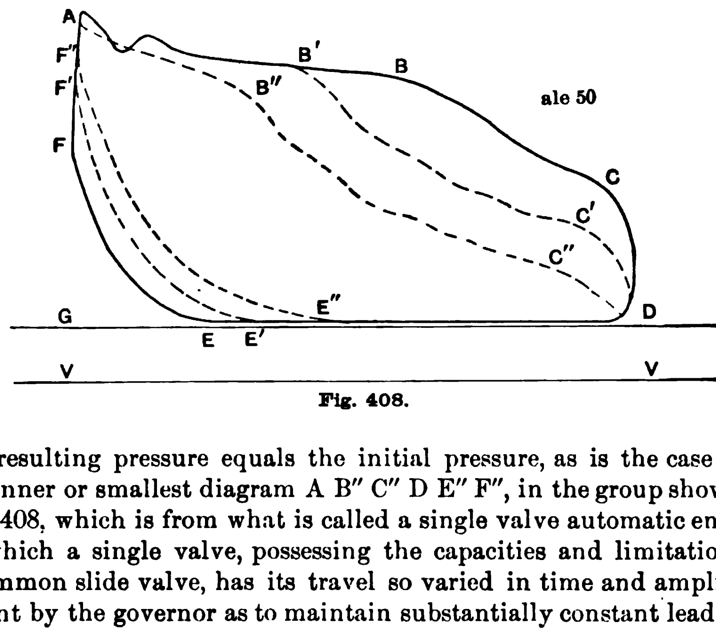


Fig. 408.

the resulting pressure equals the initial pressure, as is the case with the inner or smallest diagram A B'' C'' D E'' F'', in the group shown in Fig. 408, which is from what is called a single valve automatic engine, in which a single valve, possessing the capacities and limitations of a common slide valve, has its travel so varied in time and amplitude extent by the governor as to maintain substantially constant lead with variable cut off, but at the expense of considerable variation in the points of exhaust and exhaust closure, as is clearly shown, when such engines are run empty, the exhaust closure takes place near mid stroke, hence the need for large clearance, 7 to 10 per cent. being about the range in practice. Most of the high speed engines are of this class.

THE ATMOSPHERIC LINE

The atmospheric line G is drawn while the indicator cock is so placed as to admit the atmosphere below the piston. It may be drawn by applying the pencil while the drum is in motion, but it is preferable

to make it larger than the intended diagram, and that is done by pulling the cord by hand.

The foregoing list includes all that the indicator itself can produce, but for further analysis two more lines must be considered.

THE VACUUM LINE.

The vacuum line V V is located below and parallel with the atmospheric line, at a distance from it corresponding with the known or estimated pressure of the atmosphere at the time and place of taking it. In the absence of such knowledge, however, it is customary to assume the sea level mean, 14.7 pounds per square inch, as has been done with the diagrams shown herewith; but since the pressure diminishes over one-half of a pound, .533, for each 1000 feet of elevation above sea level, there are many cities and large areas of country of sufficient altitude to require material allowance for it. For instance, Denver, Col., being situated at 5267 feet altitude, the mean atmospheric pressure is but 11.9 pounds; at the altitude of the lakes, except Ontario, it is 14.4 pounds; and at Altoona and Johnstown, Pa., and a large portion of Ohio, it is 14 pounds.

A barometer which has been adjusted for weather readings at a particular locality, is of no use for determining atmospheric pressure, but if made or adjusted to read according to altitude, whether in terms of inches of mercury, feet altitude, or pounds atmospheric pressure, it is exactly what is required, and such can be had.

THE CLEARANCE LINE.

The clearance line V A (Fig. 409) is drawn perpendicular to the atmospheric line, at a distance from the initial end of the diagram and bears the same relation to its length that the volume of clearance space does to the piston displacement. When the construction permits, the clearance is most readily and accurately measured by filling the space with water. The volume so ascertained in cubic inches is multiplied by 100, and divided by the cubic inches of the piston displacement; the quotient will give the clearance as usually expressed, namely: the per cent. of the displacement.

When clearance can not be ascertained, and it becomes necessary to estimate it, the following facts should be borne in mind:

The same volume of clearance makes a larger per cent. of the displacement when the stroke is short than when it is long.

THE CLEARANCE OF BUCKEYE ENGINES.

The clearance of Buckeye engines is somewhat less than that of any other build, there being but two ports, and these short and direct.

They are built in three styles: Style "A," having the stroke a little less than twice the bore; style "B," stroke slightly shorter in proportion, and style "C," stroke one and one-third times the bore and less. Their clearances are about 2, 3½ and 5 to 6 per cent., respectively, the latter figure being that of one or two sizes whose strokes are but little more than the bore. The large port area required for high speed also raises the percentage in the latter class.

FOUR-PORT ENGINES.

Four-port engines, of which the Corliss is a type, have 3 to 4 per cent. clearance. The smallness of the figures being due mainly to their relatively long strokes.

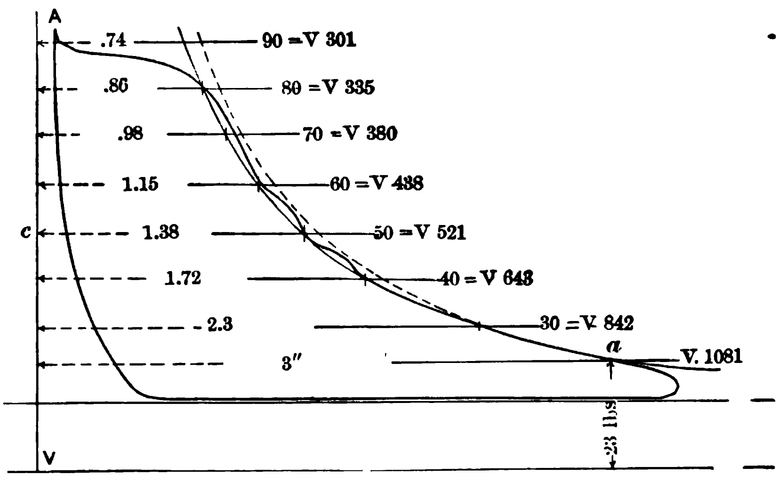


Fig. 409.

SINGLE VALVE HIGH-SPEED ENGINES.

Single valve high-speed engines have, as before stated, about 7 to 10 per cent. clearance, so as to accommodate the excessive compression which accompanies early cut off, moreover, with steam ports of the length required by the central position of the valve, and of the capacity required for high speed, such clearance is unavoidable.

THEORETICAL EXPANSION CURVES.

In the construction of theoretical expansion and compression curves by which to test the correctness of actual ones, it is customary to assume the unqualified correctness of the Mariotte law, that *the volume of elastic fluids is inversely as the pressure*—that is to say, if steam at a certain pressure be allowed to expand to double its original volume its pressure will fall one-half, if expanded to three times its orig-

inal volume its pressure will fall to one-third, and so on for all degrees of expansion—so that if the different volumes be multiplied by their resultant pressures a constant product will be obtained. But the law is true only for a constant temperature at all pressures, whereas, giving no transmission of heat to or from the fluid during the expansion, the sensible temperature will fall with the pressure, causing the latter to be materially less after expansion than the Mariotte law calls for. Thus, if steam at 90 pounds total pressure, above vacuum, be expanded 6 times, the pressure should fall, according to the Mariotte law, to 15 pounds, whereas, owing to the accompanying fall of temperature from 320° Fahrenheit to 213° Fahrenheit, the pressure will fall to a small fraction over 13 pounds, or nearly 2 pounds more in consequence of the fall of temperature.

A curve constructed in accordance with the Mariotte law, as above defined, is called the isothermal curve, the term signifying *same temperature.*

A curve in which the effect of temperature is correctly allowed for, is called the adiabatic curve, the term signifying *no transmission*, since if no heat be transmitted to or from the fluid while undergoing a change of pressure its temperature will change directly with the pressure.

Since the expansion of steam in an engine cylinder takes place too quickly to allow any considerable amount of heat to be transmitted to it from the walls of the cylinder, it would seem that the expansion curve should be more nearly adiabatic than isothermal, and under favorable conditions, such is sometimes the case. The conditions required are: engine not too small, not less than 150 horse power; tight-fitting valves and piston, and cylinder and steam pipe well protected with non-conducting covering, or otherwise; steam sufficiently superheated to reach the cylinder without entire loss of superheat.

Generally, and particularly with small engines, it will be found that actual expansion curves are still more incorrect than the isothermal curve in the same direction, namely: undue terminal pressure; so that the isothermal curve is sufficiently correct for all practical purposes, and when an actual curve agrees closely with it, the curve is considered practically perfect, but since more perfect curves are sometimes met with, it is well for the student of steam engineering to understand the nature and extent of the inaccuracy of the isothermal curve. Such an understanding will result from a study of the following processes for their delineation:

THE MATHEMATICAL METHOD OF DRAWING THE ISOTHERMAL CURVE.

Having drawn the vacuum and clearance lines as correctly as possible, select a point in the expansion curve at which it is desired that it shall coincide with the theoretical curve. The point may be near

the point of cut off or exhaust, or anywhere between the two. If near the cut off it will show what should be the resulting terminal, if near the exhaust it will show where the point of cut off should be, and if later than actual it will show how much more work should have been done with existing consumption of steam; or a point may be found somewhere in the curve which will give a theoretical diagram showing equal work, in which case, if theoretical terminal is lower than actual, it will show how much less the consumption should have been for the the same work. In the present case we will select a point near the terminal at *a* (Fig. 409), where the total pressure is 23 pounds (40 scale), and the distance from the clearance line *c* (which represents the volume due to that pressure) is 3 inches. Multiplying these together we have 69, which on our present assumption will be the constant product of pressure and volume throughout the curve, therefore, we may divide it by any number of other volumes—distance from clearance line—to obtain resultant pressures, or other pressures to obtain resultant volumes. Choosing the latter plan, we draw horizontal lines at 30, 40, 50 up to 90 pounds, or as high as the diagram requires. The constant 69, divided by each of these pressures, gives the volume number marked on each line, which is the distance in inches from the clearance line to the desired point in the curve. The points so obtained may be connected by a continuous curve, as shown in Fig. 409, or if sufficiently numerous they will serve the purpose without connecting.

THE GEOMETRIC METHOD FOR DETERMINING THE THEORETICAL CURVE.

The geometric method for determining the theoretical curve requires the card to be pinned to a board accurately squared, and a small draughting tee square and a hard, finely-pointed pencil.

The same diagram as shown in Fig. 409 is shown in Fig. 410, and the point of coincidence at C is the same as before.

From C a vertical line is drawn upward to B, where it joins the horizontal line A B, the exact height of which above the diagram is immaterial. It may represent the boiler pressure when known, otherwise it may be drawn as most convenient. From the intersection of these lines at B, draw the diagonal line B V, to the intersection of the vacuum line and clearance lines. From C draw line to D parallel with the atmospheric line cutting B V at D, and from their point of intersection draw vertical line D E, which locates the theoretic point of cut off at E, assuming A E to be the theoretical initial pressure line. Then from points F G H I J K draw perpendicular lines downward far enough to pass the probable path of the desired curve, and from same points draw diagonal lines F V, G V, H V, etc., cutting the line E D at *f*

g h, etc., from which points draw horizontal lines to intersect the vertical lines under F G H, etc., which points of intersection will be in the desired curve.

Points F G H I, etc., may be located at random, but should be closer together near E than they need be near B.

All vertical lines should be parallel to the clearance line A V, and all horizontal lines parallel to the atmospheric line L M.

The two curves agree substantially and would do so exactly with exact calculations and measurements in the first case (Fig. 409) and equally exact location of lines and intersections in the second case (Fig. 410.)

They show the expansion curve a little more correct than the isothermal curve.

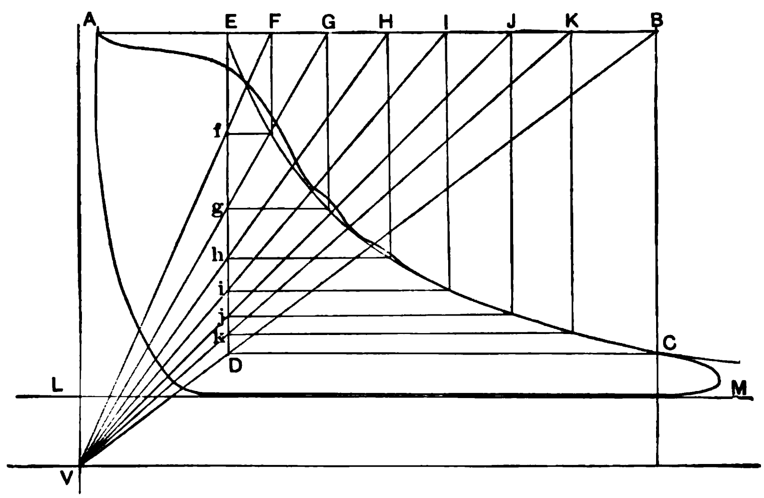


Fig. 410.

THE ADIABATIC CURVE.

The most that can be done on this subject is to give the student a tolerable correct idea of the difference between this curve and the other, since the basis of all rules for tracing it must be the volumes of saturated steam of different pressures experimentally determined, something that has not yet been accomplished with absolute certainty. Rankin's formula that "the pressure varies nearly as the reciprocal of the tenth power of the ninth root of the space occupied," is not correct unless the assumed volumes on which it is based are correct. In view of these considerations we prefer to give a process which is *correct in itself*, the correctness of the result obtained depend-

ing upon that of the assumed volumes, which being merely factors may be changed for others which may be known or believed to be more correct.

TO TRACE THE ADIABATIC CURVE.

The process that will be here given is like that first given for tracing the isothermal curve, except that instead of treating the different volumes inversely to the pressures, they are obtained from the tables of properties of steam given elsewhere in this book.

Referring to the diagram, as shown in Fig. 409, suppose the volume of steam at 23 pounds pressure, as at the selected starting point *a*, to be 1081 times that of water of same weight (this volume is represented as before by the distance of the clearance line 3″), then, as 1081 is to 3″, so is the volume of any other pressure to the length of the line representing it, measuring from the clearance line, hence the following rule:

RULE.—Divide the length of the line 3″ representing the volume of the pressure at the starting point by that volume, 1081, *and multiply the quotient by the volumes of any number of other pressures, and the products will be the lengths of the lines representing such volumes.*

Thus, beginning at the highest pressure, the quotient as above begins with .002775, and the volume of 90 pounds being 301 the length of that line will be .002775 times 301 equal .835275″, or .83″ instead of .74″, as per the isothermal curve. In like manner the other lines are found to be as follows:

For 80 pounds .93″ instead of 86″.
For 70 pounds 1.06″ instead of 98″.
For 60 pounds 1.21″ instead of 115″.
For 50 pounds 1.44″ instead of 1.38″.
For 40 pounds 1.78″ instead of 1.72″.
For 30 pounds 2.33″ instead of 2.3″.

Setting off these measurements on the pressure lines the points in the desired curve are obtained through which is drawn the dotted curve.

The test shows that the actual curve is between the two, which is a very good showing for so small an engine 12″ x 20″. But in all such cases it should be borne in mind that a certain amount of piston leakage would cause a deceptive appearance of correct expansion.

Whether any such leakage existed in the present case is not known with certainty, but the correctness of the compression curve negatives the supposition.

ANOTHER METHOD.

Move the clearance line toward the diagram, a distance equal to about one-fortieth of the length of the diagram, about in contact in the present case, and construct an isothermal curve based on such a changed clearance. The result will not be exactly the same as the foregoing, but it will be about the same at the two extremes, and not noticeably different throughout.

THEORETICAL CLEARANCE.

Select two points in the expansion or compression curve, as A B or *a b*, as shown in Fig. 411, from which points draw horizontal and vertical lines, forming the parallelogram A C D B, or *a c d b*. Through angles C D or *c d* draw a diagonal line, continuing it until it intersects

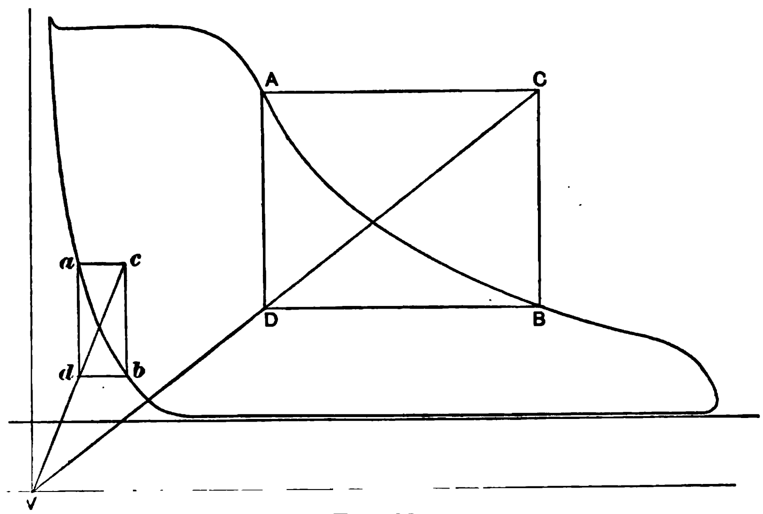


Fig. 411.

the vacuum line at V, from which point draw a vertical line which will be the theoretical clearance line. Sometimes the clearance thus indicated will be less than the actual, and sometimes more. When the conditions are such that an adiabatic curve should be produced, it should show about 2½ per cent. less than actual.

HORSE POWER CALCULATIONS.

To find the horse power of an engine it will be found most convenient to first find its horse power constant, that is the horse power due to one revolution per minute and one pound mean effective press-

ure. This constant number multiplied by the number of revolutions per minute will give the horse power per pound mean effective pressure, and the product multiplied by the mean effective pressure will give the horse power for that pressure.

RULE.—Multiply the area of the piston in square inches by twice the length of the stroke in feet, and divide the product by 33,000; *the quotient will give the constant.*

Example.—Let 22 inches equal diameter of cylinder.
Let .7854 equal a constant.
Let 2.75 feet equal length of stroke.
Let 2 equal the number of strokes for each revolution.
Let 33,000 equal a constant.

Then we have:

$$\frac{22 \times 22 \times .7854 \times 2.75 \times 2}{33{,}000} = .0633+ \text{ A constant.}$$

Entire accuracy, however, requires allowance for the area of the piston rod, which, assuming it to be one-sixth the diameter of the piston, or one thirty-sixth of the area of the piston, and that it affects one stroke only of the two, the area of piston may be diminished by $\frac{1}{72}$ of itself making it 374.8, instead of 380.13 and giving .0624 as a constant instead of .0633.

THE MEAN EFFECTIVE PRESSURE.

The mean effective pressure is most conveniently measured with the planimeter, but as complete instructions in the use of this instrument will be found in a preceding chapter, we will here confine ourselves to the method for obtaining the mean effective pressure by ordinates. It is customary to draw ten vertical lines, called ordinates, across the diagram, as shown in Fig. 412, and measuring and recording the pressure shown at each, adding the pressures and dividing the total by ten to obtain the mean effective pressure.

If the spaces are all equal the measurements for pressure should be taken in the centers of the spaces, but it is better and more accurate to begin and end with spaces one-half the width of the rest, so that measurements taken on the ordinates will stand for the centers of equal spaces. But the following method of measuring the ordinates is preferable to the one described, and, if done with reasonable care, will be more accurate.

Cut a narrow strip of paper and apply it to each ordinate in succession, setting off its length accurately with a sharp hard pencil, thus

getting the aggregate length of all of them. This length $9\frac{1}{32}$ inches in diagram, as shown in Fig. 412, is then multiplied by the scale and divided by 10. Thus:

$$\frac{9.03 \times 40}{10} = 36.12 \text{ pounds. Mean effective pressure.}$$

When negative ordinates are encountered, as Nos. 6, 7, 8, 9 and 10, in the friction diagram, shown by the dotted curve in Fig. 412, they must be measured in the opposite direction on the paper from the rest, so as to diminish the aggregate length of the positive ordinates, for the reason that they represent a partial vacuum behind the piston.

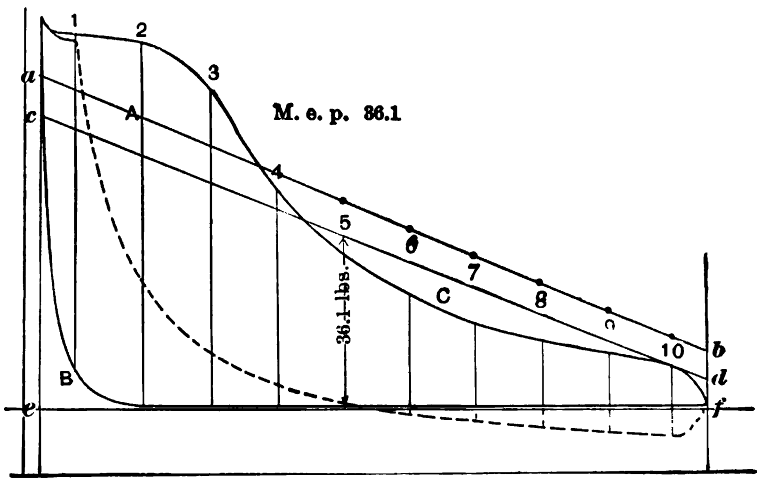


Fig. 412.

SPACING ORDINATES.

A parallel ruler is made for the purpose, and which is very convenient, but it can not be compared with the planimeter. In the absence, however, of any such instrument, the spaces may be laid off with dividers; or, having drawn perpendiculars touching the ends, a rule, graduated to sixteenths, twelfths, tenths and eighths may be applied in a more or less diagonal position, varying the angle until a distance between the verticals is formed which contains a number of one of the above subdivisions which is divisible by 10, 20, or 30, when nine full spaces with half spaces at the ends can be pointed off with a pencil and the ordinates drawn through the points. Thus line *a b* (Fig. 412) $3\frac{3}{4}$ inches or $\frac{60}{16}$ long; hence $\frac{3}{16}$ of an inch at each end and $\frac{3}{8}$ of an inch for each of the rest gives the desired spacing.

A QUICK ESTIMATE OF MEAN EFFECTIVE PRESSURE.

Draw line *c d*, as shown in Fig. 412, touching the exhaust end and cutting off an area A equal to B and C, as nearly as can be estimated by inspection, when the pressure at the middle (ordinate No. 5) will be the mean effective pressure, since the middle width of the wedge *c d e f* is its width.

This method is not applicable, with any degree of accuracy, to diagrams representing the extreme and sudden changes due to light load and high pressure; but under favorable conditions, a little practice, with results tested by other methods, will reduce the margin of error to within one or two pounds. First attempts will generally give too high a result, since area A looks larger than B and C when they are equal.

COMPOUND ENGINES.

The entire calculations of each cylinder may be made separately, and the results in indicated horse power (I. H. P.) added together; but for certain purposes, it is best to add to the mean effective pressure (M. E. P.) of one cylinder the equivalent of the other or others, thus: Suppose there are two cylinders—one high and the other low pressure—and it is desired to reduce the mean effective pressure of the low pressure cylinder to an equivalent pressure for the high pressure cylinder.

RULE.—Multiply the square of the diameter of the low pressure cylinder by the mean effective pressure of that cylinder, then divide the product by the square of the diameter of the high pressure cylinder, and the quotient will give the desired equivalent; which, if added to the mean effective pressure of the high pressure cylinder, the sum will give a pressure that, if acting in the high pressure cylinder alone, would equal the energy of both high and low pressure cylinders; and the same process may be applied to the other cylinders of triple and quadruple expansion engines, and any one of the cylinders may be selected as the exponent of all.

Example.—Let 21 inches equal diameter of low pressure cylinder.
Let 20 pounds equal mean effective pressure of low pressure cylinder.
Let 12 inches equal diameter of high pressure cylinder.
Let 40 pounds equal mean effective pressure of high pressure cylinder.

Then we have:

$$40+\left(\frac{21^2\times 20}{12^2}\right)=101.25 \text{ lbs.}$$

Performing the operation in the ordinary way, we have:

$$
\begin{array}{r}
21 \\
21 \\
\hline
21 \\
42 \\
\hline
441 \\
20 \\
\hline
\end{array}
$$

$12 \times 12 = 144)\ 8820\ (61.25$ lbs. Equivalent to 61.25 pounds mean effective pressure in the high pressure cylinder.

$$
\begin{array}{r}
864 \\
\hline
180 \\
144 \\
\hline
360 \\
288 \\
\hline
720 \\
720 \\
\hline
\end{array}
$$

Adding the equivalent mean effective pressure to the actual mean effective pressure of the high pressure cylinder, we have:

40	lbs.	M. E. P. high pressure cylinder.
61.25	lbs.	Equivalent M. E. P.
101.25	lbs.	M. E. P. of both cylinders combined in one.

THEORETICAL WATER CONSUMPTION.

The theoretical or actual economy of an engine is best expressed in terms of the number of pounds of water used per indicated horse power per hour with dry steam. In all thorough tests of engine performance both theoretical and actual consumption are tabulated—the former as "water accounted for by the indicator", and the latter as the "remainder found by subtracting from the water actually fed to the boiler" the water found to be present in the steam, as spray or supersaturation, by the calorimeter test.

The water accounted for by the indicator will seldom exceed 90 per cent. and may be considerably below that, according to the conditions as to load, pressure, protection against radiation, leakages, etc.

With simple non-condensing automatic engines the best theoretical economy is shown with the highest pressure, and such a point of cut off as will give a terminal pressure of one or two pounds above the atmosphere. But since the unindicated loss—from leakage and condensation—is much more nearly a constant *quantity* than a constant *percentage* of steam used, it follows that increase of load diminishes the *percentage* of unindicated loss, and for that reason improves the

actual economy up to the point where the increased consumption due to the increased terminal pressure begins to overbalance the gain from diminished percentage of unindicated loss. Hence, the propriety of rating engines on the basis of a mean effective pressure in excess of that which would show the best theoretical economy, at about one-fourth cut off instead of one-fifth or one-sixth.

TO CALCULATE THE THEORETICAL CONSUMPTION OF WATER.

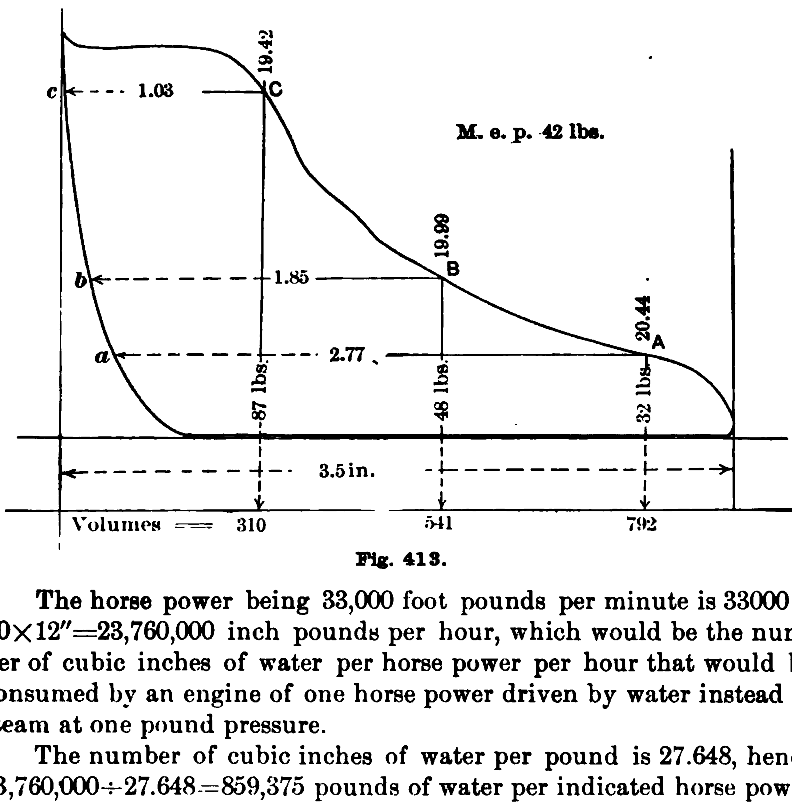


Fig. 413.

The horse power being 33,000 foot pounds per minute is 33000× 60×12″=23,760,000 inch pounds per hour, which would be the number of cubic inches of water per horse power per hour that would be consumed by an engine of one horse power driven by water instead of steam at one pound pressure.

The number of cubic inches of water per pound is 27.648, hence 23,760,000÷27.648=859,375 pounds of water per indicated horse power per hour such an engine would consume.

With more than one pound pressure the consumption of water would be proportionately less; and with steam instead of water it would be as much less as the steam at the pressure used is lighter than water, hence the following process will give as correct results as any other:

Select one or more points in the expansion curve at which it is desired to ascertain the number of pounds of water per indicated horse power per hour, accounted for by the indicator, as shown in the diagram of Fig. 413.

Three points have been selected in the diagram: A, at or near the exhaust; B, near the middle of the curve, and C, at or near the cut off. If one only be selected, it should be A; if two, A and C, which are enough for all practical purposes.

The following table contains the data:

MEAN EFFECTIVE PRESSURE, 42 POUNDS.

	Total Pressure.	Volume.	Length of Consumption Line.
A	32 pounds.	792 times water.	2.77 inches.
B	48 "	541 " "	1.85 "
C	87 "	310 " "	1.03 "

By the term "consumption line" is meant a line the length of which represents the amount of steam consumed at a given pressure as compared with the length of the diagram, thus: Line A *a*, as compared with the whole length of the diagram, represents the amount of steam consumed at 32 pounds; B *b*, the amount at 48 pounds; and C *c*, the amount at 87 pounds. The lines begin at the point of given pressure in the curve and end at the compression curve, running parallel with the atmospheric line. The reason for stopping at the compression curve is that the pressure at the selected point in the expansion curve, after further expansion and release, is restored by compression at the point in the return stroke corresponding with the point in the compression curve where the consumption line meets it; so that the line being shortened by compression in proportion to its amount, the process correctly credits the gain due to compression, while the loss of power from compression is debited in the calculation of the mean effective pressure. The greater the volume of clearance, the more gradual the rise of compression pressure and the longer the consumption line; and it will be seen that the process also debits the loss due to exhausting from the clearance space.

RULE.—Divide the constant obtained according to the rule previously given by the mean effective pressure, then divide the quotient by the length of the diagram, in inches; then multiply the last quotient by the length of each of the consumption lines, and divide the product by their respective volumes.

Thus:

$$\frac{859375}{42}=20461.3095+$$

Dividing the quotient by the length of the diagram (3.5 inches), we have:

$$\frac{20461.3095}{3.5}=5846.08+$$

This quotient serves for all of the selected points. Omitting the decimals, we have the indicated rate:

$$\text{For point A,} \quad \frac{5846 \times 2.77}{792} = 20.44+$$

$$\text{For point B,} \quad \frac{5846 \times 1.85}{541} = 19.99+$$

$$\text{For point C,} \quad \frac{5846 \times 1.03}{310} = 19.42+$$

It is evident that with perfectly adiabatic expansion and compression curves, according to the table of volumes in the chapter on "Evaporative Tests," the above calculations, if based on the same table, would give the same result for all points in the curve. Such discrepancies in the results, as shown, will be found in practice, though the degree of discrepancy will sometimes be greater, and sometimes the indicated rate will be less, in the middle of the curve than anywhere else; particularly so with small engines, owing to considerable condensation of steam by the cylinder walls soon after cut off, and the re-evaporation of the resulting water toward the end of the stroke.

This process of condensation and re-evaporation will always go on, to a greater or less extent, unless prevented by the use of superheated steam or steam-jacketing the cylinder, since the temperature of the cylinder walls will be lower than that of the initial steam, but higher than that of the terminal steam. The process of condensation frequently gets credit for more curve distortion than is really caused by it, for the reason that cut-off valve leakage is frequently responsible for most of it, especially when the terminal pressure is considerably in excess of that due to the cut off, and the condition of steam generation and transmission are such as to give fairly dry steam. In such cases there will be little or no water to re-evaporate, except that resulting from condensation in the cylinder, and that alone could only raise the terminal pressure to what it would have been without condensation; so that the presence of terminal pressure in excess indicates either wet steam or leaky valves, or both.

SLIDE VALVE PERFORMANCE.

Engineers should understand that the several functions of the common slide valve can not all be equalized, for the reason that there is unequal piston velocity during the two halves of the piston's travel, caused by the virtual shortening of the connecting rod by being deflected from the center line, as it is at all times except when the crank is on its dead centers. When the crank pin is at the middle of

its path between centers the piston is not at the middle of its travel, but it is nearer the shaft by an amount which may be found by subtracting the square root of the difference between the squares of the length of the crank and connection from the length of the connection. Thus, suppose the crank to be 12 inches and the connection 60 inches long, then we have:

$$\sqrt{(60\times 60)-(12\times 12)}=58.78+ \text{ inches.}$$

The distance from center of shaft to center of cross head wrist when the crank is midway of its travel between centers. Hence,

$$60-58.78=1.22 \text{ inches.}$$

The distance the piston is nearer the shaft than to the middle of its stroke.

But in practice there is no necessity for making this calculation; all that need be known in regard to the possibilities and limitations of the slide valve and its relation to the piston movement distortion can be shown geometrically by the process here given.

TO DETERMINE THE POINTS OF CUT OFF AND EXHAUST CLOSURE.

Draw a circle of any convenient diameter, which for large engines may be equal to the valve travel, and for small engines may be larger—in some cases equal to the piston travel. The larger the circle the more accurate the results. This circle represents both valve and piston travel as occasion requires. In the illustration (Fig. 414) it represents an assumed stroke of 24 inches, so that when it represents piston travel it is on a scale of $1\frac{1}{2}$ inches to the foot, and $\frac{1}{8}$ inch represents 1 inch. When the two travels are not so conveniently related to each other a scale may be constructed by which to make measurements for stroke events, as shown in Fig. 414, and also for valve measurements, when, as may sometimes be most convenient, it is drawn larger than the valve travel.

Through the circle draw diameter line A B, and extend it beyond the circle far enough for subsequent operations.

Now suppose a valve with $\frac{5}{8}$ inch steam lap and $\frac{3}{16}$ inch exhaust lap, to determine the points of cut off and exhaust closure due to 3 inch travel and $\frac{1}{16}$ inch lead at each end.

At the intersections of the circle with line A B, draw small circles, with given leads ($\frac{1}{16}$ inch) for radius, and from center C draw two circles, one with given exhaust lap for radius, and one with given steam lap for radius. Draw lines *a b* and *a′ b′*, tangent to the lead circles and the steam lap circle, and parallel to each other. From points of inter-

section *b b′* of these lines, with the circle draw arcs *b c* and *b′ c′* with radius equal to the length of the connection, according to the scale by which the scale equals the stroke of the piston, and with the center on the prolongation of the line A B. In the illustration the connection is assumed to be five times the length of the crank, or $7\frac{1}{2}$ inches by the scale.

The intersections *c c′* of these arcs, with the lines A B locates the points of cut off—A *c* 20 inches, and B *c′* about $18\frac{1}{16}$ inches.

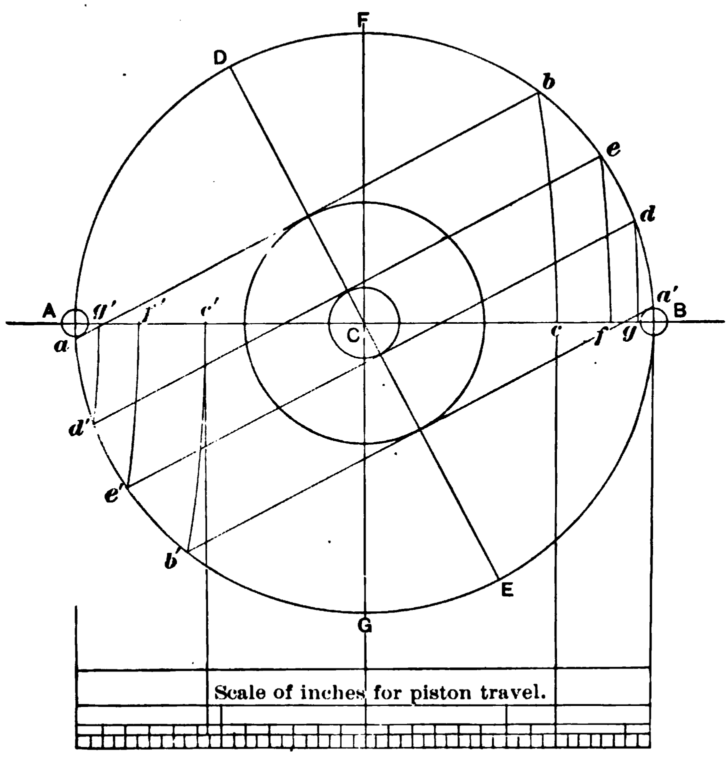


Fig. 414.

Also draw lines *d e* and *e′ d′*, tangent to the exhaust lap circle and parallel to *a b* and *a′ b′*. Draw arcs *e f e′ f′* and *d′ g′* with radius and center on line A B, locating points of exhaust closure *f f′* and exhaust *g g′*. These events are also unequal, compression A *f′* being $2\frac{1}{2}$ inches and B *f′* $1\frac{3}{4}$ inches.

Line D E is drawn at right angles to *a b* and the others, and F G at right angles to A B, hence, the angle D F or E G is the angular advance of the eccentric.

TO DETERMINE THE CONSEQUENCES TO OTHER EVENTS OF SETTING THE SAME VALVE FOR EQUAL CUT OFF.

It becomes evident by inspection that cut-off equalization must be effected at the expense of considerable lead inequality, and that the greatest lead must be at B, which corresponds with the crank end, so called, of the cylinder.

We have now no use for lead and lap circle, as these must be unequal and subject to the requirements in hand.

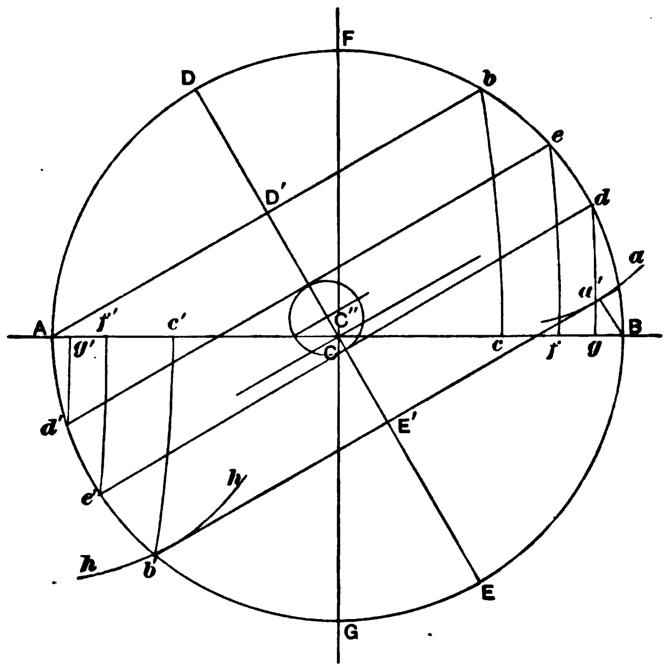


Fig. 415.

The valve and its travel being assumed to be unchangeable, we can not locate the cuts at will, but they must be determined as follows:

The sum of the laps, equal or unequal, remains unchanged. This sum, of steam lap, being 1¼ inches, with that amount as radius draw an arc *h h*, as shown in Fig. 415, from A as the center, and tangent to which draw a short portion of line *b′ a*, placing the ruler above B a distance equal to the estimated lead that will be required with none at A to accomplish the requirement, but make the line long enough only to cut the circle at *b′*, from which point draw the arc *b′ c′*, locating one point of cut off at *c′*. Set off A *c* equal to B *c′*, or B *c* equal to A *c′*; draw arc *c b* and line *b* A. Complete line *b′ a*, making it parallel to A *b*,

but tangent to arc *h h*, giving the lead required at *a'* B. If this lead has been considerably misestimated when establishing point *b'*, that point will be slightly changed, and strict accuracy will require a corresponding correction of the resulting points of cut off; but the points of tangency with arc *h h* and of intersection with the circle are so close together that the error may be neglected in practice.

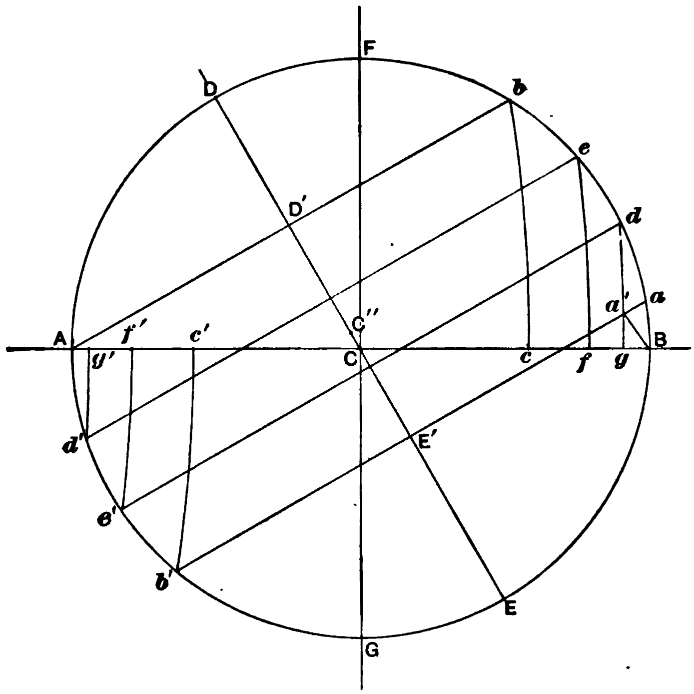


Fig. 416.

TO LOCATE EXHAUST CLOSURE.

To locate exhaust closure draw D E (Fig. 415) through center and at right angle to the other lines, as shown in Fig. 414. Bisect D' E', obtaining point C'', from which set off each way on line D E, the former exhaust laps ($\frac{3}{16}$ inch), or draw a $\frac{3}{8}$ inch circle, as shown; tangent to which draw lines *d' e* and *e' d* parallel to A *b;* then draw arcs *d' g' e' f' d g* and *e f*, locating exhausts at *g'* and *g*, and exhaust closures at *f* and *f'*. The latter will be unequal, being about $2\frac{5}{16}$ inches at end A, and $2\frac{1}{16}$ inches at B. Exhaust inequality is of no great consequence.

It is now evident that it is not advisable to entirely equalize cuts at the cost of such great lead inequality, although if clearance, load and steam pressure were such that compression pressure would just about

reach the initial pressure, the excessive lead at crank end would cause no loss of good running qualities, and the compression might be equalized by changes of exhaust laps, now about $\frac{1}{16}$ and $\frac{5}{16}$ inch; but first let it be determined what the result will be by equalizing the compression by adjustment alone, without change in the proportions of the valve.

The compressions obtained in Fig. 415, being slightly less than $2\frac{1}{2}$ inches at end A, and slightly more at B, set off at f' and f equal compressions of $2\frac{1}{2}$ inches, and draw arcs $f\ e$ and $f'\ e'$, as shown in Fig. 416. Next, set off exhaust g slightly less than corresponding one in Fig. 414, and draw arc $g\ d$ and line $d\ e'$, obtaining the angle. Then line $e\ d'$ parallel to $d\ e'$ and arc $d'\ g'$ locates the other exhaust at g'. Obtain point C″, as before, and set off steam laps C″ D′ and C″ E′ on line D E. Through points D′ and E′ draw lines parallel to the others, and arc $b\ c$ and $b'\ c'$, locating the cut offs at c and c', $19\frac{1}{4}$ inches and very nearly 19 inches respectively. Owing to the use of a little guess work in locating exhaust at g the sum of the exhaust laps slightly exceeds $\frac{3}{8}$ inch, which, if entire accuracy is sought, would call for a slightly earlier exhaust point at g, which would change the angle of the parallel lines and bring the exhaust lap lines $d'\ e$ and $d\ e'$ closer together.

The inaccuracy is left uncorrected, however, to show that the "cut and try" process is to some extent required when lead equality is departed from and fixed valve proportions are adhered to.

TO FIND THE VALVE LAPS REQUIRED FOR GIVEN RESULTS.

Having learned that entire equality of all events is impossible, it now remains to find the best possible compromise among the several distortions shown in the illustrations.

The excessive lead inequality required to equalize the cuts is objectionable, if for no other reason than that many engineers would consider it a mistake or accidental mal-adjustment that should be corrected. Moreover, since unequal cuts are neither detrimental to the economy or running qualities of the engine, we may safely take the illustration shown in Fig. 414, so far as the steam laps and leads are concerned, and change the exhaust laps so as to give equal compression of chosen amount, determining the amount with reference to the known volume of clearance, so as to compress from three to five volumes into one, according as the speed may be high or low.

The construction of Fig. 417, which will now be sufficiently understood without further explanation, shows equal leads and compressions—the former $\frac{1}{16}$ of an inch, as shown in Fig. 414, and the latter

2 inches. The cuts are at $18\frac{1}{2}$ inches and 20 inches. The exhaust laps are $\frac{1}{4}$ inch and $\frac{1}{12}$ inch, the smallest determining the compression at the end of the cylinder farthest from the shaft, where the piston moves faster than it does at the opposite end. The steam laps are both $\frac{5}{8}$ of an inch, as with Fig. 414. Angular advance of eccentric 28 degrees, as shown in Fig. 417.

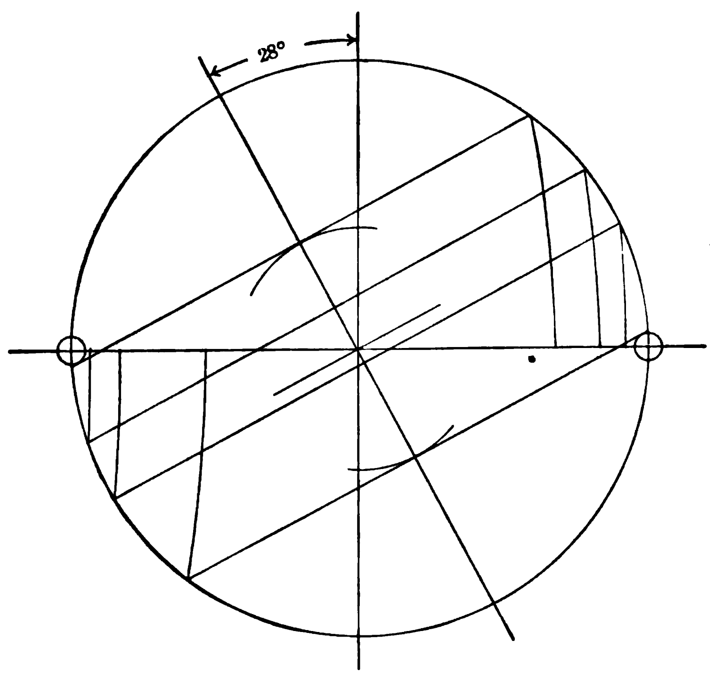


Fig. 417.

TO DETERMINE SLIDE-VALVE FUNCTIONS—CONDENSED FORM.

Having presented the subject of slide-valve functions in detail for the benefit of the student, it will now be given in condensed form. The circle A C B D, as shown in Fig. 418, may be of any convenient diameter; but the larger it is the more accurate will be the results obtained. It represents the strokes of both the valve and piston, and if not drawn to agree with either, scales may be constructed, by one of which its diameter will equal the stroke of the piston, and by the other the stroke of the valve. In the present case it will be assumed to equal the valve travel, so that only a piston travel scale will be needed; and assuming a piston travel of 24 inches, a scale rule of $1\frac{1}{2}$ inches per foot will answer.

Line A B is drawn through the center and extended to the length required outside of the circle.

Lines *a b*, *c d*, *a' b'*, *c' d'* and *e e'* are drawn parallel to each other, and *e e'*, shown broken, through the center.

Arcs *b f'*, *d g'*, *c' h'*, *b' f*, *d' g* and *c h* are each drawn with a radius equal to the connecting rod of the engine according to the scale by which the diameter of the circle equals the piston travel, and from

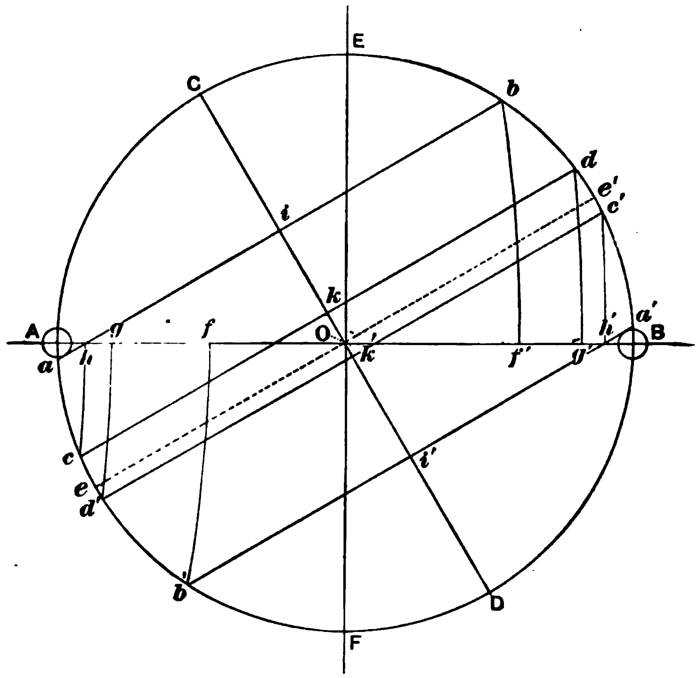


Fig. 418.

centers located on the prolongation of the line A B. They are drawn from the intersections of lines *a b*, *c d*, etc., with the circle to the line A B.

By the scale according to which the circle represents the valve travel—the scale in the present case—the radii of the small circles at A and B are the leads, the centers of the circles being at the intersections of the line A B with the main circle, and lines *a b* and *a' b'* being drawn tangent to them.

Distance C *i'* and *i* D are the maximum port openings.

Distance *i* O and *i'* O are the steam laps, O being the center of the circle.

K O and K′ O are the exhaust laps, the smallest K′ O being at the crankward end of the cylinder to equalize compression.

The points of cut off are f and f'. They are unequal in consequence of the slower movement of the piston during the crankward half of its stroke than during the other half, due to the virtual shortening of the connecting rod when deflected from the center line.

The points of the exhaust closure g g' are located at 2 inches from the ends of the stroke, in the present case.

The points of the exhaust fall at h and h' as a result of all the other conditions.

Angle C E is the angular advance of the eccentric, line C D being at right angles with the parallel lines, a b, c d, etc., and E F at right angles with line A B.

It is obvious that many variations may be made on the above layout, for example: by making more lead at B and less at A the cut offs $f f'$ may be made equal. If less compression be desired distances A g and g' B may be diminished accordingly, from which new points arcs g d and g' d' and lines d c and d' c' are redrawn, showing the required diminution of exhaust laps K O and K′ O. Different angles for C E may also be chosen, showing effects on the several events, all of which may be done correctly by the student by following these instructions.

COMPOUND ENGINES.

RATIO OF CYLINDERS.

The best ratio of capacity of cylinders depends on the range of pressures, that is, on the ratio of initial pressure to pressure at release. For example, suppose the initial to be 115 pounds, and the steam is to be expanded until its total pressure is only 11 pounds; which would be an expansion of steam from one volume to about 10½ volumes, and the ratio of expansion would therefore be 10½. If the steam be cut off in the small cylinder at $\frac{3}{10}$ of the stroke, then the total capacity of the large cylinder would be 10½ times $\frac{3}{10}$ of the small cylinder, for $\frac{3}{10}$ of the small cylinder represents the amount of steam admitted at 115 pounds pressure, which finally fills the large cylinder at 11 pounds pressure. But 10½ times $\frac{3}{10}$ is about 3⅛; therefore the ratio of cylinders in this case would be 3⅛ to 1. This is about the ratio used in most of the Buckeye Compound Condensing Engines, and while the figures employed are approximated the result obtained is accurate enough for all practical purposes. A larger ratio of cylinders and a greater expansion would produce greater economy in case the load was carefully adjusted to the engine; but as that is not practical except where engines are specially designed to meet certain conditions, a ratio of cylinders of 3 to 1, for 10 to 10½ expansions of steam will be found good practice.

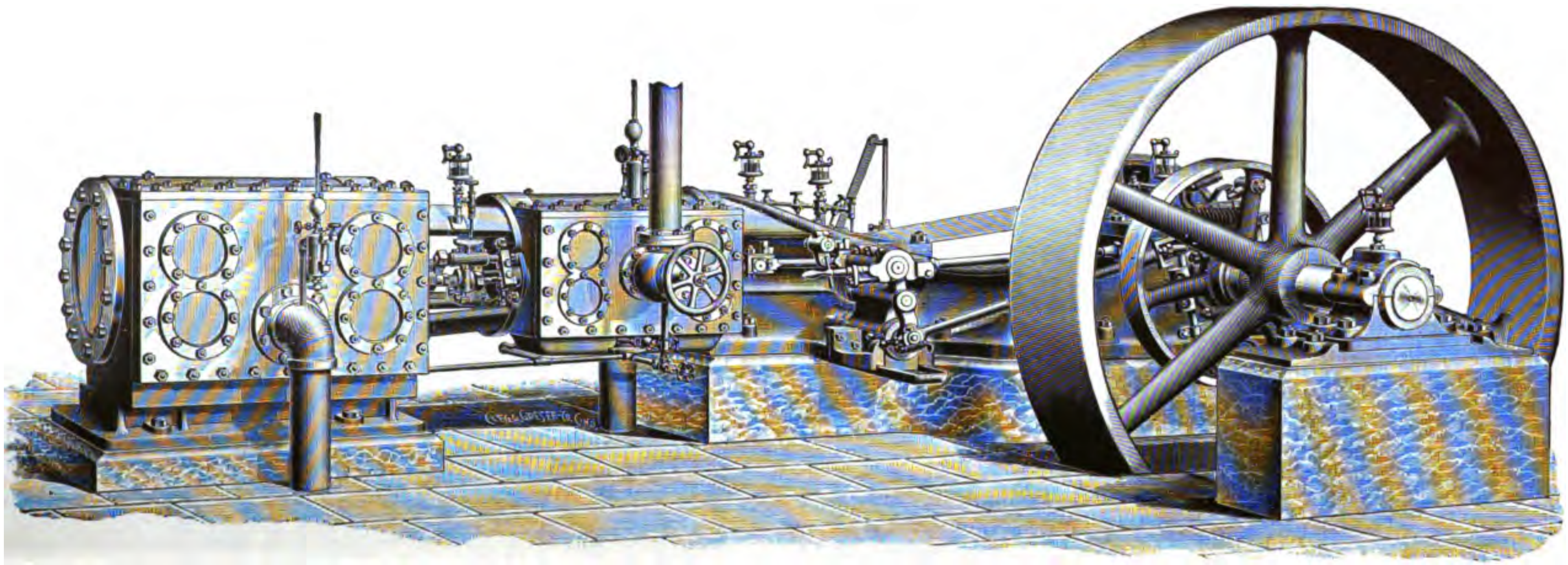


Fig. 419.

BUCKEYE TANDEM COMPOUND ENGINE—REAR VIEW.

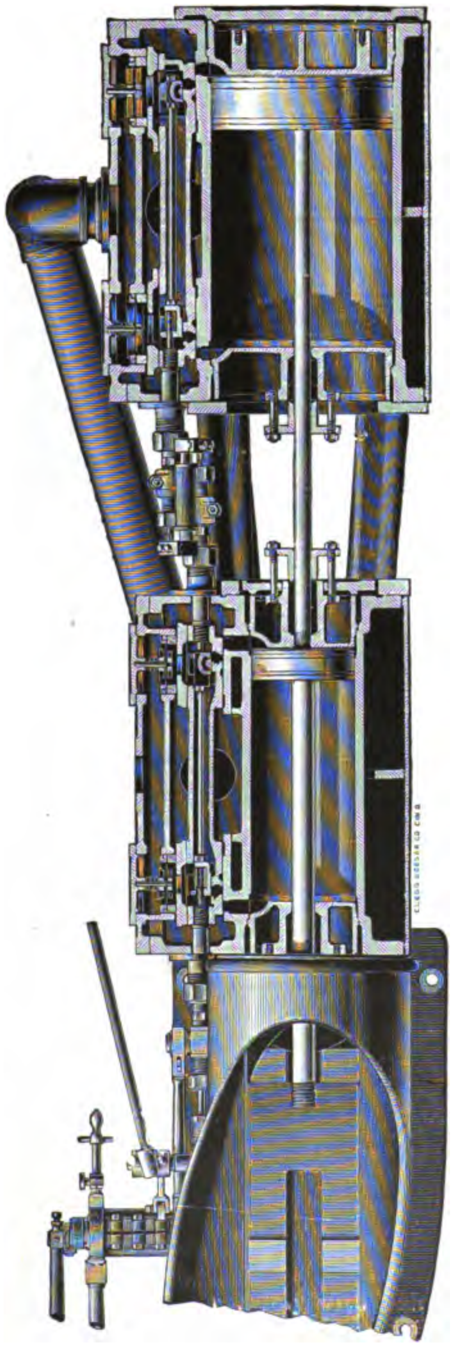


Fig. 420.
SECTIONAL PLANS OF CYLINDERS AND VALVE GEAR.

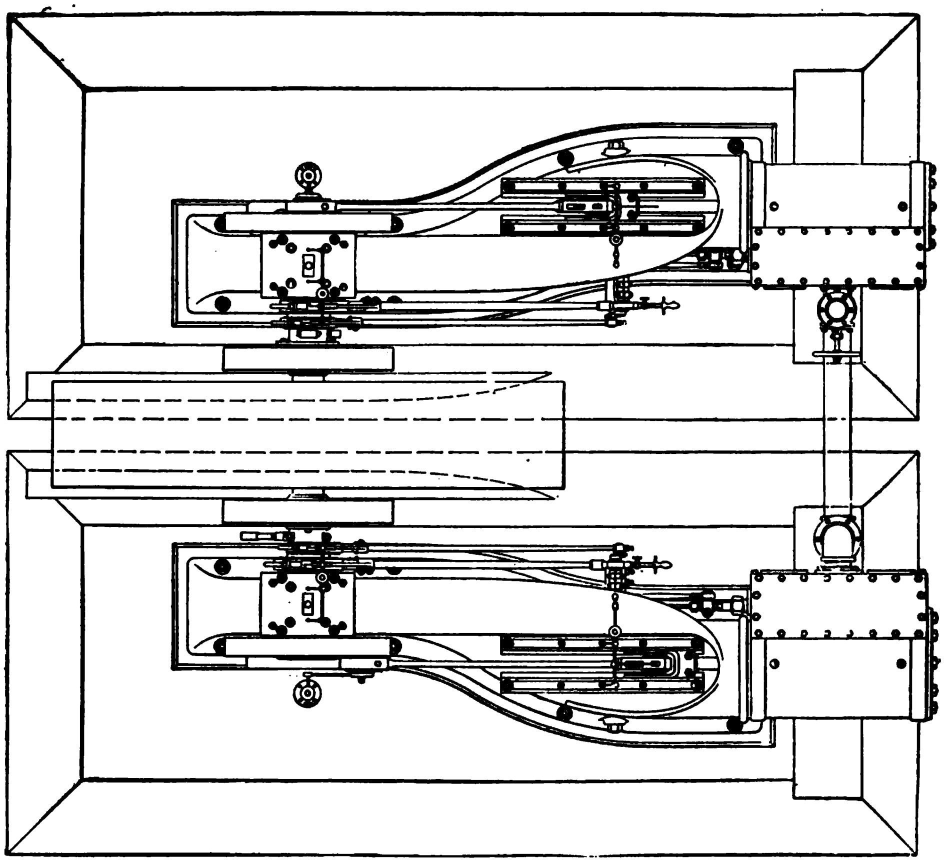


Fig. 421.

PLANS OF CROSS COMPOUND ENGINE.

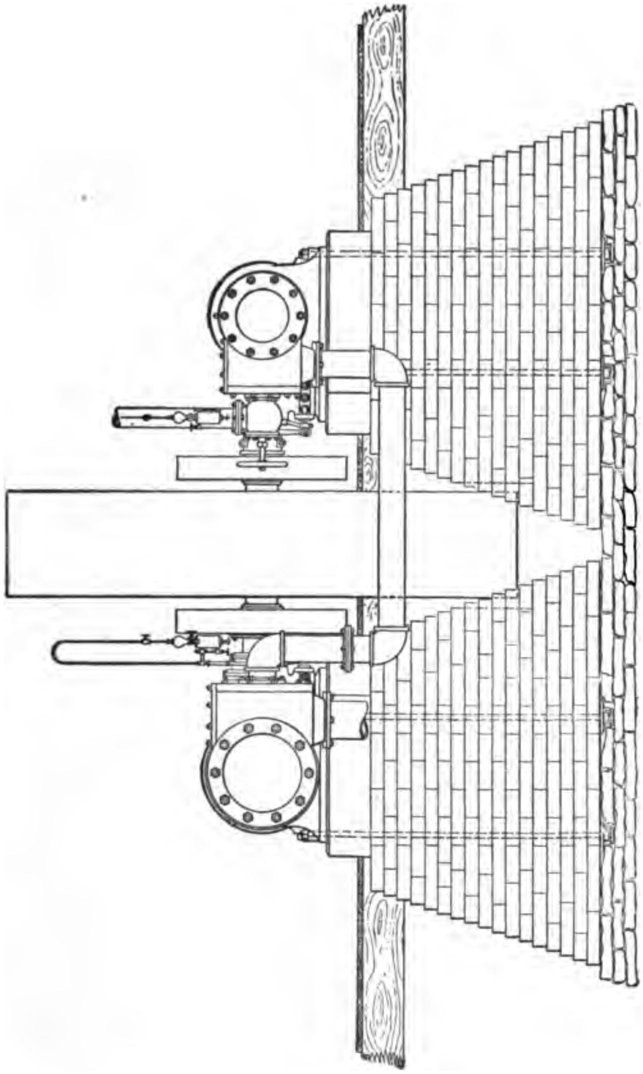


Fig. 422.

END ELEVATION OF CROSS COMPOUND ENGINE.

CHAPTER XXIII.

THE PORTER-ALLEN ENGINE.

This engine presents features in its valve motion distinctly peculiar to itself, and differs materially from others herein presented. Its central feature is a link, actuated by a single eccentric, from which separate and independent movements are given to the admission and exhaust valve

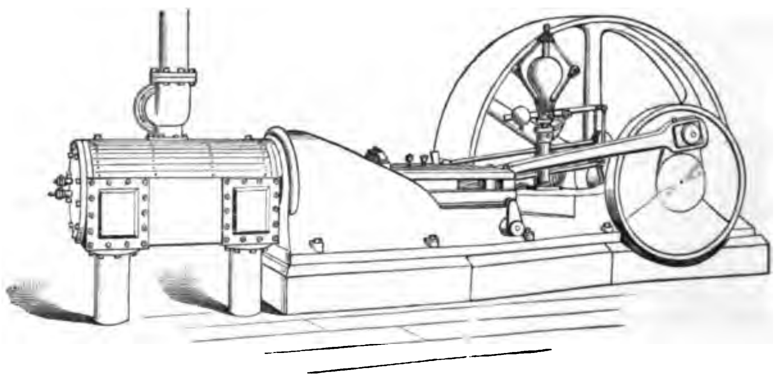


Fig. 423.

THE ECCENTRIC.

The eccentric is placed on the shaft in the same position with the crank, and can not be altered from that position. The lead of the valves is adjusted by other means. The first requirement in this system is that the crank and the eccentric shall have coincident movements, and so shall arrive on their dead points, or lines of centers, simultaneously. To insure the permanence of the eccentric in its correct position, it is formed in one piece with the shaft, and its low side is brought down to the surface of it, as shown in Figs. 425 and 426.

THE LINK.

The construction of the link is also shown in Figs. 425 and 426. It is of the form known as the stationary link, and consists of a curved

arm, partly slotted, formed in one piece with the eccentric strap, and pivoted at its middle point on trunnions, which vibrate in an arc whose chord is equal to the throw of the eccentric, about a sustaining pin secured rigidly to the bed. The radius of the link is equal to the length of the first rod, by which its motion is communicated to the admission valves.

Only the upper end of the link is here shown, being the portion used in engines which are run in the forward direction; backward-running engines require the lower end, and in reversing engines, both ends are employed.

The terms "forward" and "backward" are employed here in the reverse sense of that employed in locomotive practice; that is, the forward end of a locomotive cylinder is the rear end of a stationary engine cylinder.

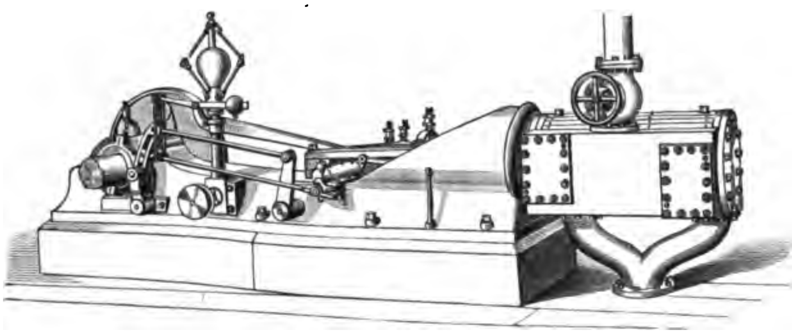


Fig 424.

In the slot is fitted a block from which the admission valves only receive their motion. This block is moved by the action of the governor, which thus varies the point of cut off. If the center of the block is brought to the center of the trunnions the port is not opened at all, except by the lead given to the valves, and this opening is closed before the piston has advanced a sensible amount. If, on the other hand, the block is brought to the end of the slot, as here represented, the steam is not cut off until the piston has reached the half stroke, which is the usual limit of the admission; although where great power is occasionally required, the steam is admitted through five-eighths of the stroke.

The exhaust valves are driven from a fixed point on the link, and have, therefore, an invariable motion. The movements of the link, at this point, are suited to this function, causing the steam, wherever it may have been cut off, to be held until near the end of the stroke,

when it receives a free and ample release, and is confined again at an earlier point of the return stroke.

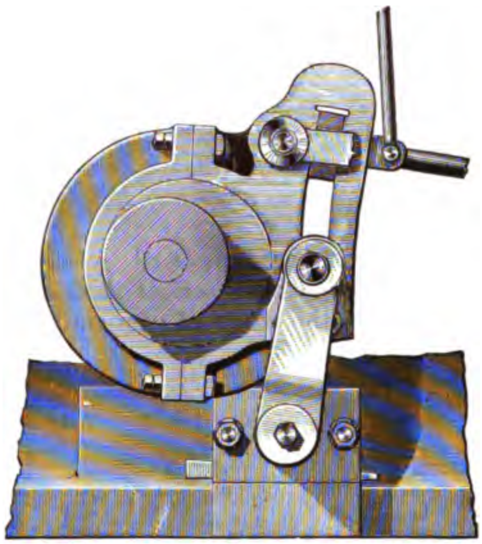


Fig. 425.

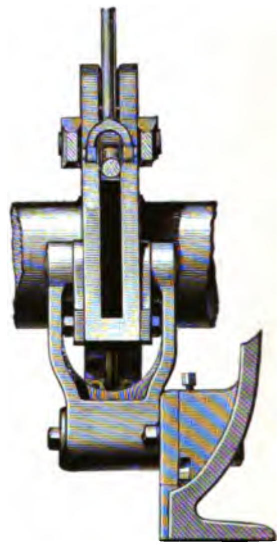


Fig. 426.

ECCENTRIC AND LINK—SIDE AND FRONT ELEVATION.

THE LINK MOTION.

An explanation of this motion is here given, reference being had to Figs. 427 and 428.

O is the center of the shaft; A B is the path of the crank; C H D I is the path of the center of the eccentric; and E K F is the arc in which the trunnions of the link vibrate, about the center G of the sustaining pin.

This arc is here divided into twelve equal parts, the terminations of which are indicated by numbers, continued through both vibrations, from one to twenty-four. The same numbers indicate corresponding points in the path of the center of the eccentric. The curved lines show the positions of the center line of the link corresponding with the positions of the eccentric and trunnions, and the progression of the numbers indicates the direction of the motion. K H and K I represent the line connecting the center of the trunnions with the center of the eccentric, at each extreme of its vibration.

This link, it will be seen, is a right angle lever, of which the above line represents one arm, which is termed its driven arm, and the curved center line of the link represents the other arm, which is termed its driving arm, the whole turning on a vibrating fulcrum.

We will at present confine our attention to the adjustment shown in Fig. 427, to show the manner in which the link imparts the movements to the admission valves. When the engine is on either dead center, the center of the trunnions of the link stand at E or F; the line connecting this center with the center of the eccentric coincides with the line of centers D F; the center line of the link stands on E M or F L, which are called the lead lines, being the arcs whose center is the joint at the other end of the valve rod; and the block can

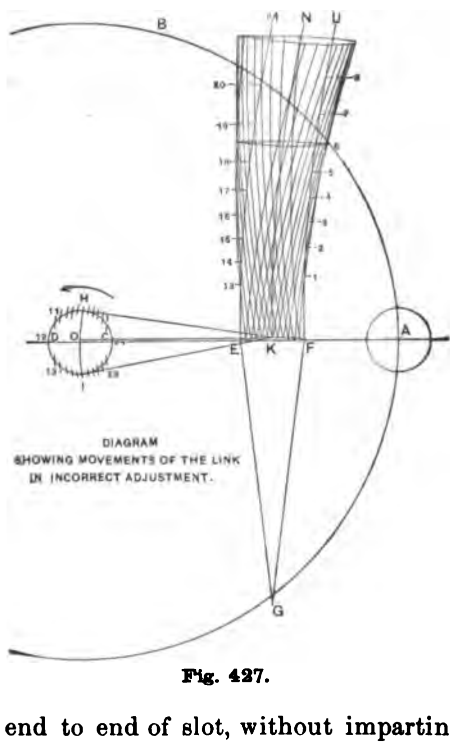


Fig. 427.

be moved from end to end of slot, without imparting any motion to the valves. The piston is then at the commencement of a stroke, and the valve has opened the port to the extent of the lead given to it. This lead is, therefore, the same, whatever may be the position of the block in the link, or, in other words, the engine has a constant lead for all points of cut off.

The peculiar motion of the link is given to it by a combination of the horizontal and vertical throws of the eccentric. The horizontal throw alone only moves the link from one to the other of the lead lines, E M, and F L, which motion only draws off the lap of the valves. The opening movement is produced by the tipping of the link alternately in the opposite directions beyond the lead lines, and these tip-

ping motions are given by the vertical throws of the eccentric. Its upward throw, from the point C, tips the link in the direction from the shaft, and opens the port at the further end of the cylinder; and its downward throw, from the point D, tips the link toward the shaft, and opens the port at the crank end of the cylinder. At the same time, its horizontal throw is drawing the valve back, and when, in this return movement, that point in the link at which the block stands crosses the lead line, the steam is cut off.

DISTINGUISHING FEATURE OF THE VALVE MOTION.

There is another distinguishing feature about the valve motion which will now be described.

The angular vibration of the connecting rod causes a considerable difference in the motion of the piston in the opposite ends of the cylinder, retarding it in the end nearest to the crank, and accelerating it in the end furthest from it. When the length of the connecting rod equals six cranks, as it does in this engine, this difference in velocity averages 20 per cent., and at the commencement and termination of the strokes reaches 40 per cent. Now the driven arm of the link is also equal to six eccentric cranks, and its angular vibrations, to K H, and K I, coincide, in degree, as well as in time, with those of the connecting rod, and so the trunnions of the link receive a motion coincident with that of the piston, and the link gives to the valves, in opening and closing their ports, different velocities, accelerated at one end of the cylinder, and retarded at the other, corresponding to the difference in the velocity of the piston. In these diagrams, therefore, the divisions of the arc represent also corresponding divisions of the stroke, and the movements of the link are correctly shown in their relation to the motion of the piston. This difference in the piston velocity, at the opposite ends of the stroke is plainly to be seen in any engine, and in these engines the same difference is to be observed in the admission valves.

Fig. 428 is a modification of the adjustment shown in Fig. 427. The adjustment of the link, shown in Fig. 427, by which the vibrations of the trunnions terminate on the line of centers, causes the link to tip beyond the lead line, further in the direction towards the shaft than in the other direction, giving a wider opening and later cut off at the crank end of the cylinder. Fig. 428 shows the adjustment by which this inequality is corrected, and equal openings are made, and the steam is cut off at identical points, on the opposite strokes, from the commencement to the middle of the stroke.

The tipping of the link in the direction from the shaft is produced by the upward movement of the point C, at the extremity of the line

C F. If the point F were at rest, the point C would describe an arc, about F as a center, opposite to the arc C H. But the point C is compelled to move in the arc C H, and thus draws after it the point F, through a distance equal to the interval between these rapidly diverging arcs, limiting the opening movement, and causing an early closing of the valve. On the other hand, the tipping of the link toward the shaft is produced by the downward movement of the point D at the extremity of the line D F. If the point E were at rest, the point D

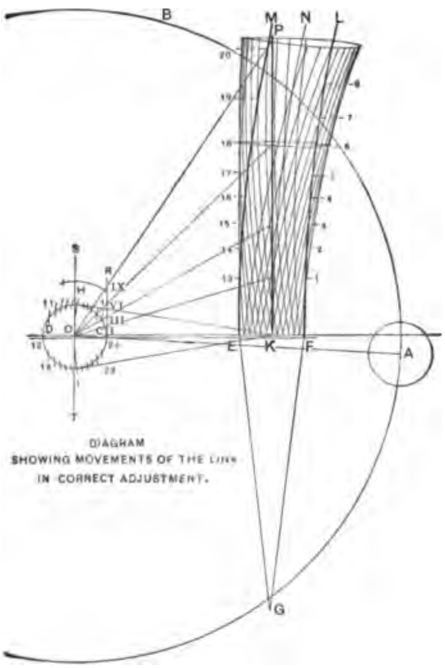


Fig. 428.

would describe an arc, about E as a center, and this arc would not be opposite to the arc D I, but would coincide with it in direction, only being drawn with a longer radius. The point D, moving in the arc D I, draws after it, therefore, the point E more slowly, permitting a wider opening and a later closing of the port.

It is the same angular vibration of the driven arm of the link, which causes the motion of the valves to differ in velocity in a degree corresponding to the difference in the velocity of the piston, that also limits the opening, and hastens the closing of that valve whose motion it accelerates, and enlarges the opening, and delays the closing, of the valve whose motion it retards.

The action of the link would be mathematically more correct if the trunnions, instead of vibrating in an arc, moved in a direct line E F (Fig. 427) along the line of centers. Then the vibration of its driven arm would be equal to the sine of angle made by it, and also by the eccentric, with the line of the centers. The lead would, as already described, be constant, and at the mid stroke the opening and cut off on the opposite strokes would be equal, since they are determined by the amount of this vibration at the point of cut off, and sine H K O equal I K O (Fig. 428). But these vibrations would be equal at the mid stroke only; as we pass backward in the stroke the vibrations for corresponding points of the opposite strokes grow unequal, and before we reach the commencement the difference in the diagrams becomes considerable.

Although by changing the line of motion of the trunnions of the link from an arc terminating on the line of centers, to this line itself, the more serious inequalities of the opposite valve movements are corrected, it still remains to remove them in the earlier part of the strokes. That is done in the following manner:

If, with a radius equal to one and a half the driven arm of the link, an arc be drawn below the line of centers, so that this line is tangent to it at O, it will be found that ordinates drawn to this arc, from points in the path of the eccentric corresponding to equal advance of the piston on its opposite strokes, as from 1 and 13, 2 and 14, and so on, are practically equal in length.

If the sustaining pin G be lowered the short distance necessary to cause the trunnions to vibrate in the arc (Fig. 428) tangent to the line of centers at K, equal valve action will be obtained on the opposite strokes, at every point of cut off, from the commencement to the middle of the stroke. The steam is cut off at the latter point when the driven and driving arms of the link are of equal length.

The link will now, however, arrive on its lead line F L, when the crank is at A (Fig. 428), before it reaches the line of centers for its forward stroke, while, on the return stroke, it will not arrive on its lead line E M, until the crank has reached a position corresponding to A, after passing the line of centers; the driven arm of the link standing then on a line below and parallel with the line of centers, as shown in Fig. 428.

This adjustment serves another purpose. It gives a difference of lead, to the amount above shown, in favor of the center where the motion of the piston is most rapid. This is an important feature, since, to obtain at high speed an equally good admission at the further end of the cylinder, it is necessary, not only that the motion of the valve shall be accelerated in the same degree with that of the piston, but also that the area of opening shall be proportionately enlarged,

and this enlarged area is, at the commencement of the stroke, where the excess of the piston velocity is the greatest, completely furnished by the greater lead.

It will be interesting to the student to observe that the movements of the link are the same as those that would be imparted by a series of eccentrics of increasing throw and diminishing angular advance. To show this:

At the point K (Fig. 428) erect K P, perpendicular to the line of centers, and through the center O draw the perpendicular S T, and draw C R tangent to the arc C H. Then, from the center O, draw a diagonal line, terminating on any point on the perpendicular K P. A portion of such a diagonal line will form a secant to the tangent C R. Four illustrations of these diagonals are given in Fig. 428, forming the secants, O I, O III, O VI and O IX.

The identity of the link with the eccentric is then exhibited, as follows:

First. The length of the secant is equal to one-half the throw of the link at the point at which the diagonal terminates.

Second. The circle being equal to the distance between the lead lines, it follows that the section of the secant beyond the circle represents the opening movement of the link at that point.

Third. The movement of that point of the link, and the opening given by it, are therefore the same that would be derived from an eccentric whose throw was equal to twice the secant, and whose advance was equal to the angle made by the secant with the perpendicular S T.

Fourth. The intersection of the secant with the circle shows the point in the revolution of the eccentric at which the full opening is found.

Fifth. This point bisects the portion of the arc that is included between the commencement of the opening and the point of cut off. The diagonal represents, therefore, the line of such larger eccentric.

The eccentric movement here exhibited assumes a connection unaffected by angular vibration.

The motion of the eccentric imparting its movement to the link, at right angles, is in direct opposition to that of eccentrics imparting similar movements directly.

STEAM AND EXHAUST VALVES

The valves are four in number, two admission and two exhaust valves, as shown in Fig. 429. The valves to the right in the illustration are the admission valves, and those to the left are the exhaust valves. There is one admission and one exhaust valve at each end of the cyl-

inder. They stand in a vertical position, and are perfectly balanced. Each of these valves opens simultaneously four passages, two on each face, for the admission and release of the steam.

The exhaust valves open and close their ports as the center line of the link crosses the line K N. The movements at the point of the link at which connection is made with the valves are shown at top of illustration. It will be seen that the opening is made while the valve is moving swiftly, and that one-half of the opening movement has been accomplished when the piston arrives at the end of its stroke. The valves are so constructed that this portion of the movement opens the full area of the port, which does not begin to be contracted again until the center line of the link has recrossed the lead lines on its return. The speed of the piston is then diminishing also, and the exhaust is not throttled at all until the port is just about to be closed.

DIFFERENTIAL VALVE MOVEMENT.

A form of the Corliss wrist motion is introduced into the connections of the admission valves, which is shown in the following plan and elevation (Figs. 431 and 432), and which effects a modification of the movements of these valves.

In this movement an arm, which is connected by a rod with the block in the link, communicates, through a rock shaft, motion to two other arms, causing them to vibrate in the same vertical plane in which the valves move. Each of these arms, alternately, rises nearly to the vertical position, while the other, at the same time, descends to and beyond its dead point.

Each, by a separate connection, imparts motion to one of the admission valves, and at the top of its vibration causes it to open and close its port swiftly, and then, descending to its idle arc, reduces the motion of the valve to an interval practically of rest.

These movements can be followed in the illustration, where the upper arm is about to move in its arc to the left, and thus, through the lower connection, to open the port at the further end of the cylinder, while the lower arm will be scarcely moving its valve at all.

In this manner the width of opening is largely increased, chiefly by a difference in the length of the levers, while, at the same time, fully one-half of the lap, or the useless motion of the valve after it has covered its port, is gotten rid of, so that smaller valves and narrower seats are employed, and, notwithstanding the greater opening movement, the total motion of the valves is very much reduced.

The valves work between opposite parallel seats, the exhaust valves are nearly and the admission valves are wholly in equilibrium. Means are provided for keeping them perfectly steam tight, the exhaust valves

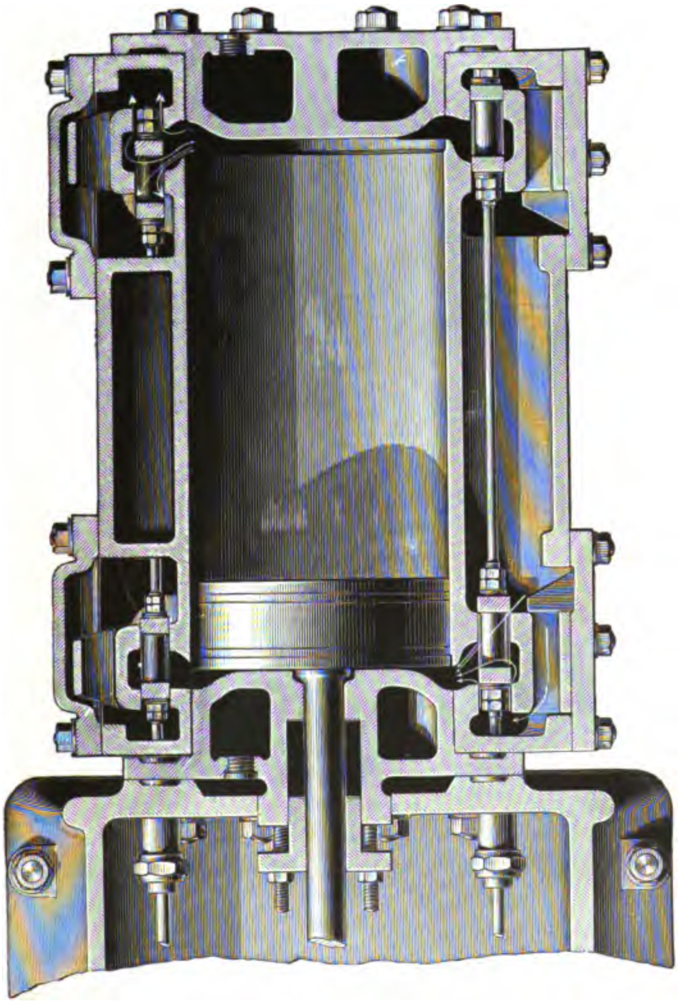


Fig. 429.

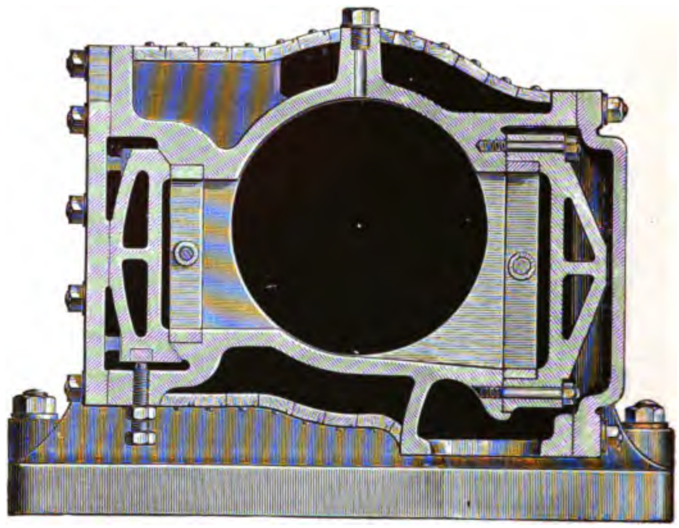


Fig. 430.

Fig. 431.

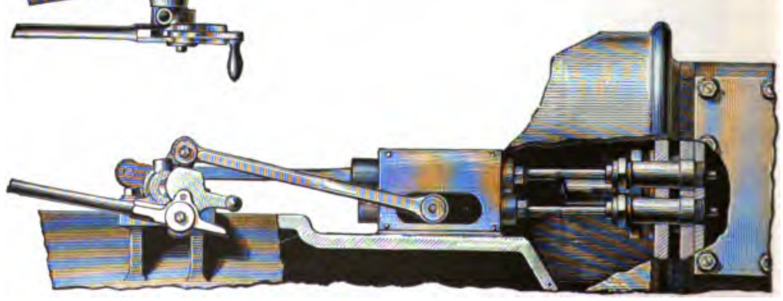


Fig. 432.

automatically and the admission valves by hand. The latter is accomplished by the employment of what is termed

ADJUSTABLE PRESSURE PLATES.

The construction of these pressure plates, and the method of adjusting them, are fully represented in Figs. 429 and 430.

On the right-hand side of the perpendicular section (Fig. 429) both admission valves are shown, working between their opposite parallel seats, one of which is formed on the cylinder, and the other on the pressure plates, the latter having cavities opposite the ports.

The valve at the upper end of the cylinder is at the extremity of its lap, while the one at the crank end has commenced to open the four passages for admission of the steam.

The vertical cross section (Fig. 430) passes through the middle of one pressure plate, and shows its form, and the means employed for its adjustment, it is made hollow, and most of the steam supplied to two of the openings passes through it. It is arched to resist the pressure of steam without deflection. It rests on two inclined supports, one above and the other below the valve. These inclines are steep, so that the plate will move freely down them under steam pressure, and also that it may be closed up to the valve with only a small vertical movement.

It is prevented from moving down these inclines by a screw passing through the bottom of the steam chest, the point of which, and the plug against which it bears, being of hardened steel.

The pressure plate is held in its correct position by projections in the chest on one side, and tongues projecting from the cover on the other, which bear against it near each end, as shown. Between these guides, it is capable of motion up and down its inclined supports, and also directly back and forth between the valve and the cover, from which it is separated by a space varying, in the different sizes, from $\frac{1}{16}$ to $\frac{1}{8}$ of an inch.

The pressure of steam is always on this plate, and tends to force it down the inclines, to rest on the valve. By means of the screw, it is forced, against the steam pressure, up the inclines, and away from the valve. This adjustment is capable of great precision, so that the valve works with entire freedom between its opposite seats, and still is steam tight.

These plates also act as relief valves. Whenever the pressure in the cylinder exceeds that in the chest, the pressure plate is instantly moved back to contact with the cover; thus affording an ample passage for the discharge of water before it can exert a dangerous strain.

To enable the student to get a still clearer idea of the construction and adjustment of the pressure plates, his attention is directed to the sectional views presented in Figs. 433, 434, 435 and 436.

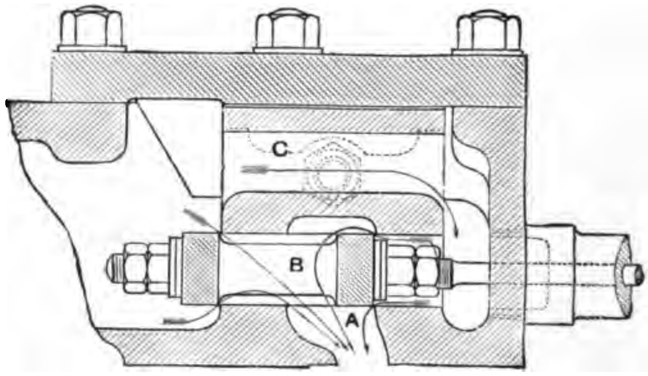


Fig. 433.

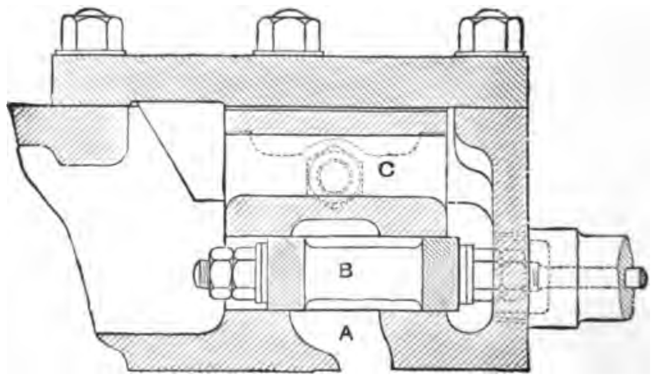


Fig. 434.

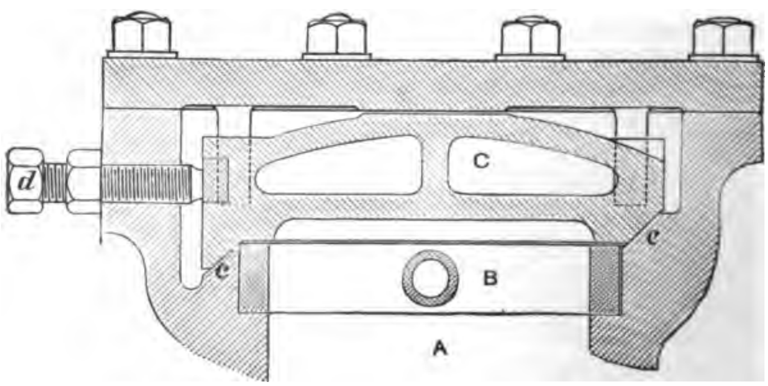


Fig. 435.

Figs. 433 and 434 are horizontal sections, through the steam chest at one end of the cylinder, showing the four openings in the valve B in Fig. 433, and also showing commencement of valve opening; and in Fig. 434 showing the valve B at the extreme point of its lap.

Figs. 435 and 436 are vertical sections showing the pressure plate C. In Fig. 435 the bolt *d* has been turned forward and forced the plate C against the inclines *c c* and away from the valve B, producing

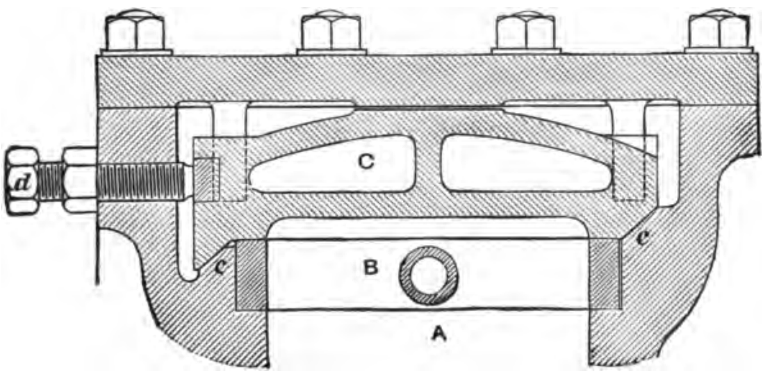


Fig. 436.

a leak. In Fig. 436 the bolt *d* has been turned back and the valve B has been let down to its proper working position. In the illustrations, A represents the steam port.

TO SET THE VALVES.

To set the admission valves place the engine on one of its dead centers; then raise the governor, bringing the center of the block between the centers of the trunnions of the link.

With the governor remaining up, set the valve that is about to open, and give it a lead of from $\frac{1}{16}$ to $\frac{3}{16}$ of an inch, according to the size of the engine and the speed at which the engine is to run. High speed engines require considerable lead.

Repeat this same operation for the admission valve on the other end of the cylinder, using the opposite dead center of the crank.

On letting the governor down, the crank remaining on the dead center, it will be seen that the valve is moved a short distance. This motion of the valve, produced by moving the block from the trunnions to the extremity of the link while the crank stands on the center, is the same in amount on either center, and takes place in the same direction; namely, toward the crank. Its effect is, therefore, to cover the port nearest to the crank, and to enlarge the opening of the

port furthest from it; so that the lead, which is equal at the earliest point of cut off, is at the crank end of the cylinder gradually diminished, and at the back end increased in the same degree as the steam follows further.

The effect of this is to equalize the opening and cut-off movements, so that, on setting the governor at any elevation whatever, and turning the engine over, the openings made and the points of cut off will be found to be identical on the opposite strokes, from the commencement up to the maximum admission. This difference in the lead is also adapted to the difference in the piston velocity at the two ends of the cylinder.

In case the indicator shows that the lead of either admission valve requires to be changed, the change can be made without opening the chest, by lengthening or shortening the stem at the socket of its guide, bearing in mind that each valve moves toward the middle of the cylinder to open its port.

The exhaust valves have an invariable motion, and they are set so as to open before the end of the stroke, enough to give ample lead, and close again when the piston is on the return stroke, early enough to effect the required compression.

All the valves are held between pairs of brass nuts, of which the inner one is flanged. These nuts must be securely locked, and should be so set upon the valve that it is free to adjust itself between the nuts while yet sufficiently tight that no lost motion exists. To avoid the consequences of a mistake, care should be taken, before closing the valve chests, to turn the engine slowly through an entire revolution, while the movements of the valves are carefully watched, so as to be sure that they have not been set so as to bring the valves or their nuts into contact with the ends of the chest at the extreme of their movements.

TO SET THE ENGINE.

The foundation should be made of hard bricks laid in cement. The bricks should be wet, and the cement washed into every course.

Time should be allowed for the cement to set before any weight is put upon it. A week, at least, is required for this purpose; four weeks would be better still. The bolts should have some play in the masonry, and the best way of insuring this is to surround each bolt with a wooden box of ½ inch material, about 16 inches long, which is drawn up as the courses are added, and removed entirely before the engine is placed on the foundation, so the bolt holes may be poured full of cement after the setting is completed.

THE BED PLATE.

This is lined in the usual manner by a line through the cylinder, which is bolted to the end of the bed, in line with the guides. In case the cylinder is not yet in place, it is represented by the bore in the head of the bed, and the line is to be continued mid way between the side rails of the lower guide bars. The guides lie in one plane, and are to be used for leveling the bed in both directions.

The base of the bed is not brought in contact with the foundation. Thin parallel packing pieces are to be placed on each side of each bolt, and under each end of the main bearing, and the bed must bear equally on all these, when the guides are level in all directions, before any strain is put on the bolts. After these have been tightened and the guides are found to be level, the broad flange of the bed is brought to a general bearing on the foundation by running with sulphur, or by calking with iron borings, wet with water, made only slightly acid with sal ammoniac.

THE SHAFT.

In placing the shaft in position three requirements must be observed:

First. That it be placed at a right angle with the axis of the cylinder.

Second. That it be level.

Third. That it lies fairly in its bearings.

The shaft is readily squared. The crank disc is finished on the shaft centers, after the pin has been set, so that if its rim on the opposite side is equally distant from its center line, the shaft is square. It is leveled by plumbing the crank disc.

When thus set it will lie fairly in the main bearing; and if the outer bearing has been correctly set, it will lie fairly in that also. This is tested by rotating the shaft entirely dry. Brightened rings will show what part of the journals have found bearings, and on lifting the shaft bright spots on the Babbitt metal will show where these bearings were.

The boxes are slightly larger than the journals, and so the latter should bear along the center of the lower box and not on the sides.

The journals of the shaft, if set as here directed, will, with ordinary lubrication, run cold from the start. Should the shaft get out of line, it may be squared by gauging between the rim of the crank disc and bases provided on the bed.

TO LINE THE ENGINE WITH THE SHAFT PLACED AT A HIGHER OR LOWER LEVEL.

We will suppose the shaft not yet in place, but to be represented by a line tightly drawn. From two points, as far apart as practicable, drop plumb lines nearly but not quite touching this line.

Then by these stretch another line parallel with the first, and at the same level as the center line of the engine, and at right angles with this, stretch another, representing this center line, and extend both each way to permanent walls, on which their terminations, when finally located, should be carefully marked, so they can at any time be reset.

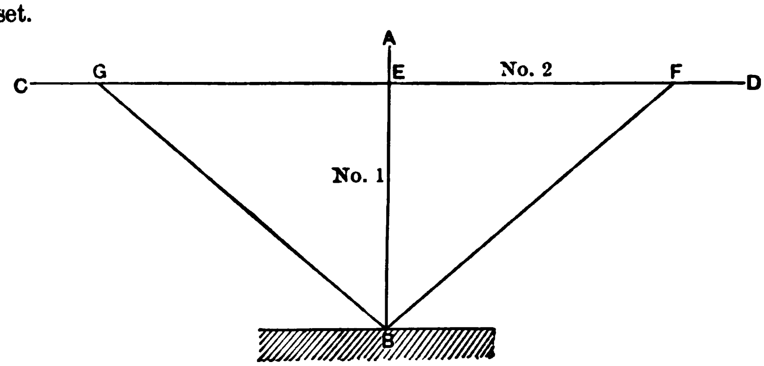


Fig. 437.

The problem is to get the latter line exactly at right angles with the former. Everything depends on the accuracy with which this right angle is determined. It is done by the method of right-angled triangles. There are two ways of applying this method:

In the first, one end of a measuring line is attached to some point of line No. 1, and its other end is taken successively to points on line No. 2 on opposite sides of their intersection, as illustrated in Fig. 437, in which A B is a portion of line No. 1, and C D of line No. 2, the direction of which is to be determined. B F and B G are the same measuring lines fixed at B, and applied to the line C D successively at the points F and G. The distance B F and B G being therefore the same, when E F is equal to E G the lines A B and C D are at right angles with each other.

In the second, application is made of the law that the square of the hypothenuse of a right-angle triangle is equal to the sum of the squares of the other two sides.

Example.—Let 5 feet equal the length of the hypothenuse of a right-angle triangle.

Let 3 feet equal the length of the base.

Let 4 feet equal the height.

Then we have:

	$5\times5=25$.	Square of the length of hypothenuse.
	$3\times3=9$.	Square of the length of the base.
	$4\times4=16$.	Square of the height.
Then:	$9+16=25$.	The sum of the squares of the other two angles.

So if the above figure E F=3, E B=4 and B F=5, the angle at E is a right angle. Any unit of measure may be used—a foot is generally the convenient one—so any multiple of these numbers may be taken, as for example, 6, 8 and 10.

Respecting the comparative advantages of these two ways, the situation will often determine which is to be preferred. In the former, the diagonal being the same line, fixed at B and brought successively to the points F and G, its length is immaterial, though generally the longer the better; and the only point to be determined is the equality of E F and E G, which may be compared with each other by marks on a rod. In the latter the proportionate lengths, 3, 4 and 5, or their multiples, must be exactly measured. It is better adapted to places where a floor is laid, and the measurement can be transferred by trammels. The result should be verified by repeating the operation on the opposite side of the intersection at E, and when so verified we have, in fact, the first process, without the additional and unnecessary trouble of determining the relative length of the lines.

Care should be taken when a measuring line is used to avoid errors from elasticity. On this account a rod is often employed. Points on the lines are best marked by tying on a white thread.

TO LINE THE ENGINE WITH A SHAFT COUPLED DIRECT.

In this case it is supposed that the engine bed and the bearings for the shaft are already approximately in position. They are leveled by a parallel straight edge and a spirit level. To line them horizontally a line must be run through the whole series of bearings, and continued to a permanent wall at each end, and its terminating points, when determined, carefully marked, as already directed. A piece of wood is tightly set in each end of each bearing, and the surface of these are painted white or chalked. Then, the middle of each piece being found by compasses, two fine lines are drawn across it, equally distant from the middle, and having a space between them a little wider than the thickness of the line. This then being strained, nearly touching those blocks, or if long, having its sag supported by them, the two marks on each block must be seen, one on each side of the line, with the line of white between them.

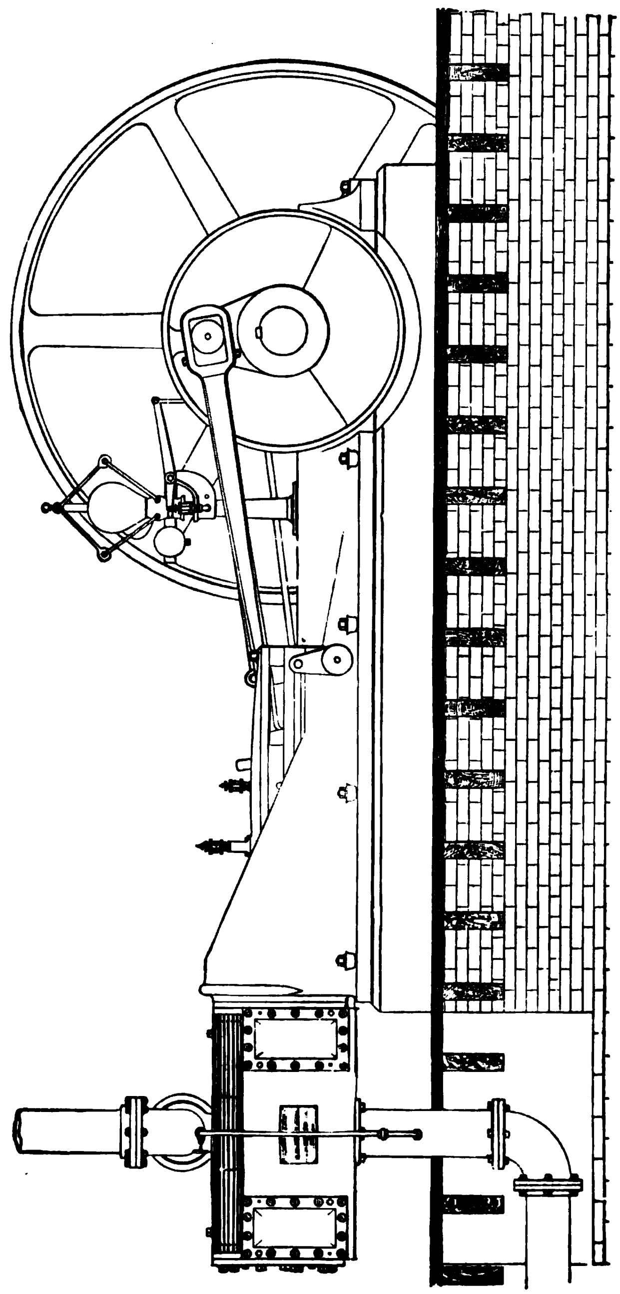


Fig. 438.
FRONT ELEVATION.

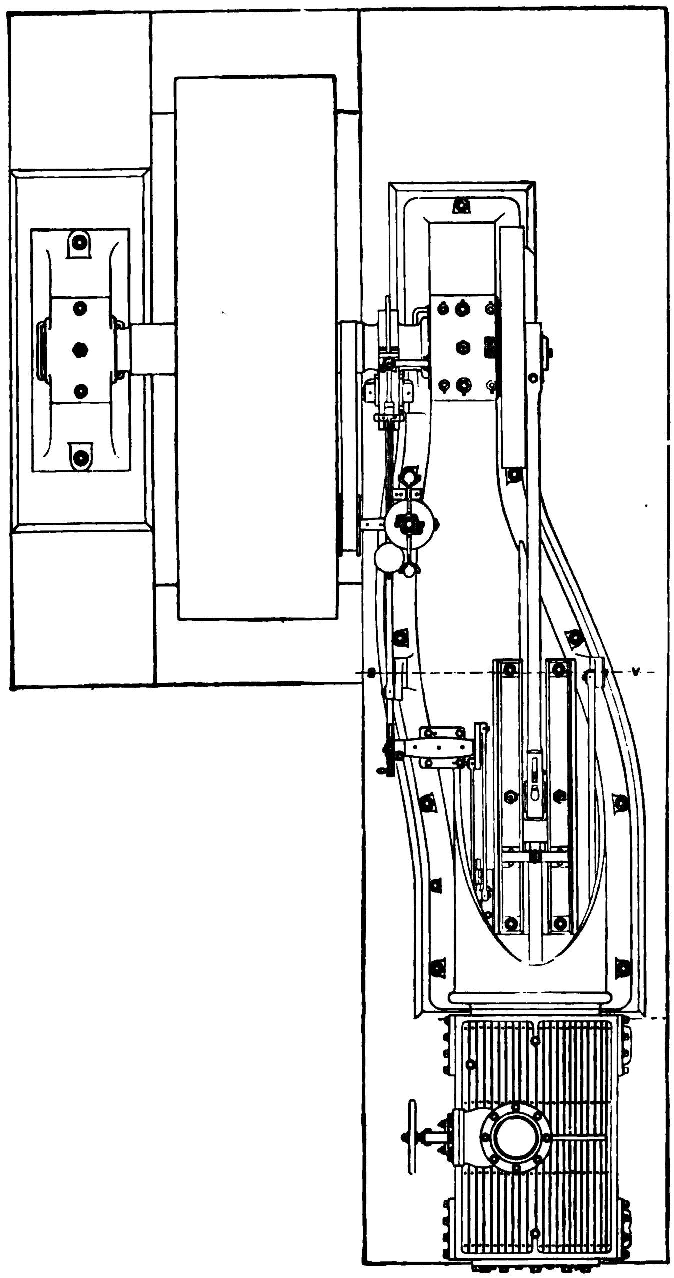


Fig. 439.
PLAN.

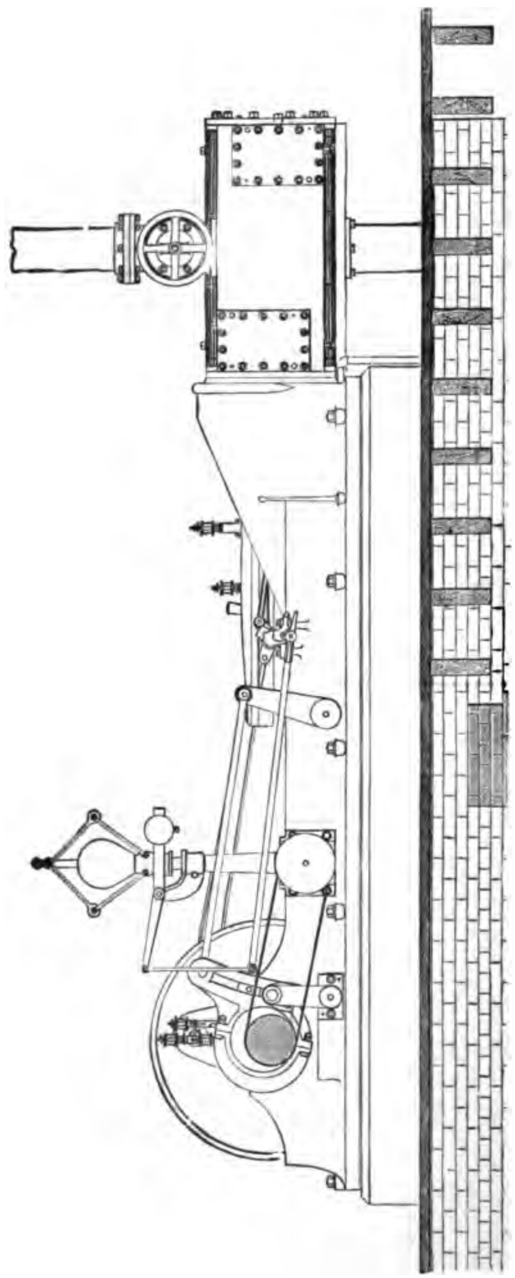


Fig. 440.
REAR ELEVATION.

ACTION OF RECIPROCATING PARTS OF STEAM ENGINES.

These parts are the piston and rod, the cross head and the connecting rod. The student, or the engineer, who is unacquainted with the forces developed in the running of these parts, and the important function which they perform, can have, at best, but a very imperfect idea of the action of high speed engines.

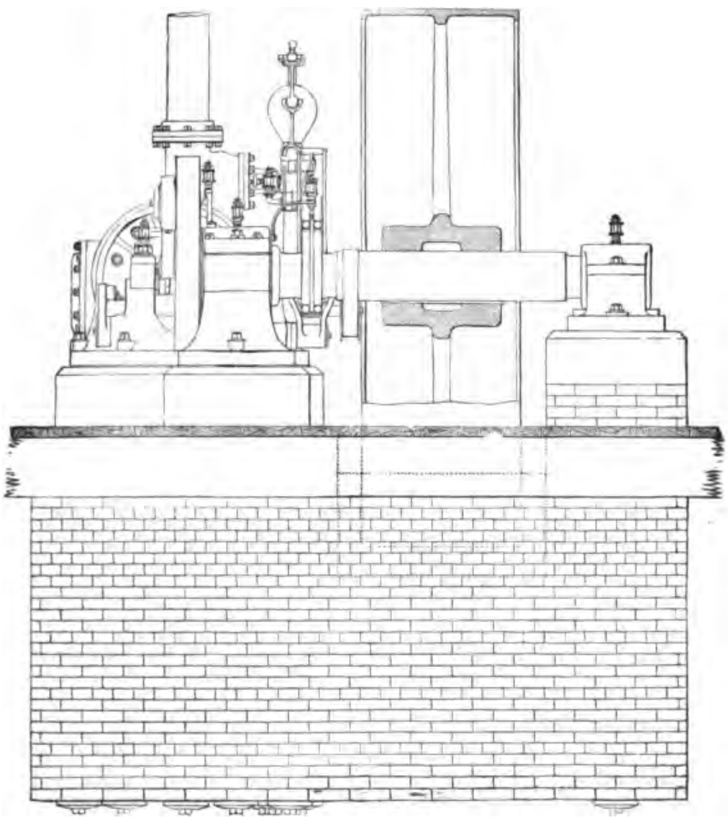


Fig. 441.
CRANK END ELEVATION—PULLEY PARTLY IN SECTION.

The great desideratum is perfect smoothness in the running of all steam engines, and to accomplish that, in high speed engines, mechanical ingenuity has encountered an obstacle which has been hard to overcome. Nevertheless it has been overcome, and this engine presents a splendid example of the triumph of mechanical ingenuity over what seemed an insurmountable obstacle for many years. The secret of it was the inertia of the reciprocating parts between the steam and

the crank. On every center they are at rest. At the middle of each stroke they have a velocity equal to that of the crank. This velocity is all imparted to them in the first half of each stroke, and taken from them in the last half. Now it is plain to be seen that these parts offer resistance to being put in motion, and when put in motion they offer resistance to being put at rest; and this resistance to changes in the amount and direction of its motion is termed its inertia, and this inertia throughout the universe of matter is a constant thing; and a certain force acting through a certain space is required to impart or to

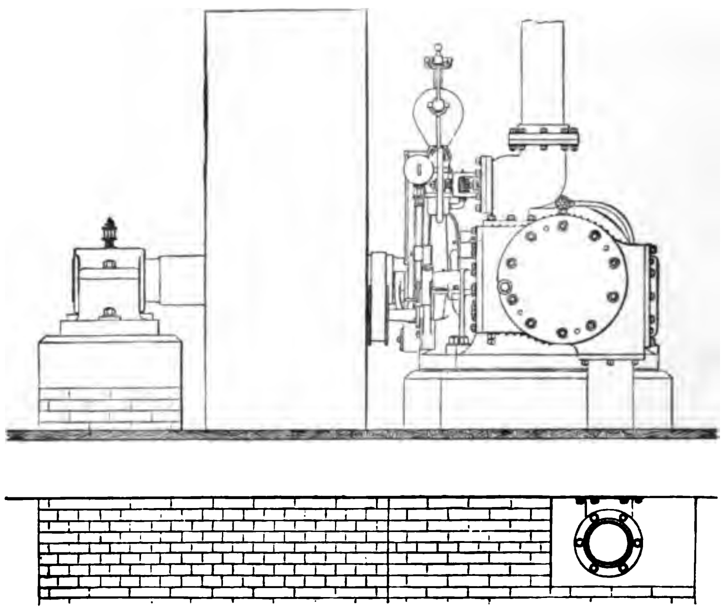


Fig. 442.

VIEW FROM CYLINDER END.

arrest a certain motion in a given body; and the force exerted varies as the square of the motion imparted or arrested within such space. Therefore, if a double velocity is imparted, it must be imparted in one-half the time, and hence requires the exertion of four times the force.

The inertia of the reciprocating parts of an engine culminates on the dead centers, those points at which their motion in one direction is finally arrested, and that in the opposite direction begins to be imparted; and there it is equal to the centrifugal force which these parts would exert if they were revolving in the path of the crank, and at every other point in their stroke it is equal to the horizontal com-

ponent of this centrifugal force, at the corresponding point in the revolution of the crank. That is, the centrifugal force is merely the inertia of a revolving body, or its resistance to a continual change in the direction of its motion.

THE UNIT OF CENTRIFUGAL FORCE.

The unit of centrifugal force, or the centrifugal force of one pound, making one revolution per minute, in a circle of one foot radius, or 2 feet in diameter, is .000341 of a pound; and this force varies directly as the weight revolving, and as the length of the crank, and as the square of the number of revolutions per minute. So the simplest computation enables us, in any case, to find what this final retarding and initial accelerating force is.

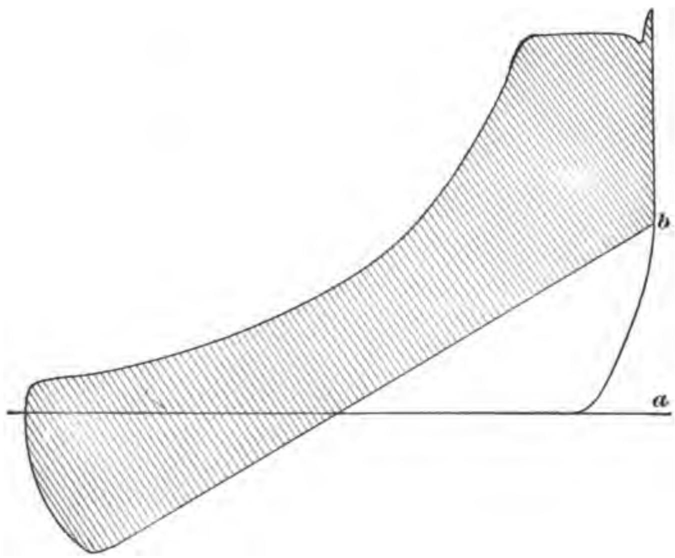


Fig. 443.

For example, in an engine of 2 feet length of crank, making 50 revolutions per minute, it is 1.705 times the weight of the reciprocating parts; for $50\times50\times2\times.000341=1.705$. That is, they are being put in motion 1.705 times as rapidly as gravity would do. We then multiply the weight of these parts by 1.705, and we have the total force; and by dividing this by the number of square inches on the piston, we have the pressure of steam per square inch that will furnish it. At the above speed it will be insignificant, probably 7 or 8 pounds per square inch; this depends, of course, upon the size of the piston. But it increases as the square of the speed; at 100 revolutions per

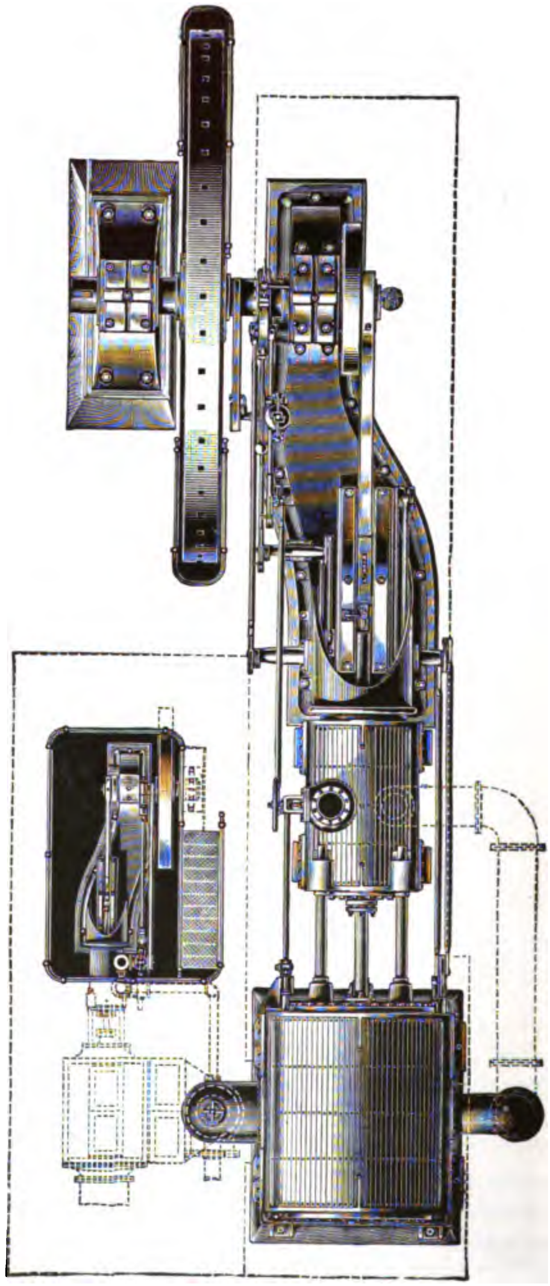


Fig. 444.
PLAN VIEW OF TANDEM COMPOUND CONDENSING ENGINE.

minute it is four times as much; for $100 \times 100 \times 2 \times .000341 = 6.82$. At 200 revolutions per minute it is sixteen times as much; for $200 \times 200 \times 2 \times .000341 = 27.28$. The first is between 7 and 8 pounds per square inch, which does not amount to very much; the second is 30 pounds per square inch, something to be considered; the third is 120 pounds per square inch, something to be avoided.

The force on the square inch having thus been found for the dead centers, we have only to measure, at the commencement of the diagram, a height representing it, as, for example, *a b* in Fig. 443, and from the point *b* to draw a diagonal, crossing the line of counter pressure at the middle of the diagram, and continue to the end; and we will have enclosed two triangles, the first representing, at every point, the force exerted to give these parts their velocity which they must have to keep up with the crank; and the second the precisely opposite force exerted to stop them.

At the mid stroke, acceleration passes insensibly into retardation. The curved end of the lower triangle in the figure shows the extent to which the motion of these parts is arrested by the compression on the opposite side of the piston, and not by the crank.

COMPOUND CONDENSING ENGINES.

Fig. 444 represents a plan of a Porter-Allen Compound Condensing Engine, and Fig. 445 represents a side elevation.

COMPOUND EXPANSION.

Economy in the use of steam is the real object of compounding, but to insure economy in a compound expansion engine provision must be made for preventing too great a condensation of the steam between the point of admission into the first, or high pressure cylinder, and the point of its discharge into the second, or low pressure cylinder. Or, if the engine is of the triple or quadruple expansion type, provision should be made to prevent too great a condensation between the point of admission of steam into the first and the point of its discharge into the last cylinder. If this is not provided against compounding can result in little, if any, benefit so far as economy is concerned.

In the construction of this engine, however, this important matter has received due attention, and all of the conditions necessary to high economy are fully met. But, in order that the student may have a clear conception of what changes take place in a steam engine cylinder, we will first consider the subject of conversion of heat into work.

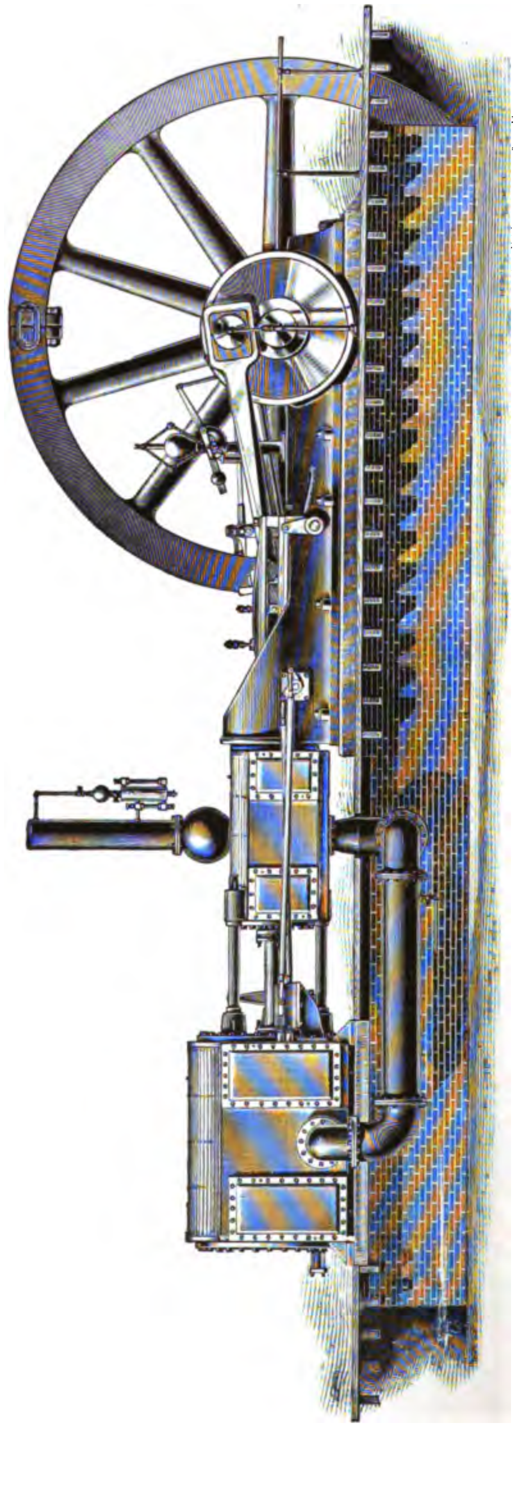


Fig. 445.
SIDE ELEVATION OF TANDEM COMPOUND ENGINE.

CONVERSION OF HEAT INTO WORK.

When the steam enters the cylinder it is as invisible as in the gauge glass, but as the piston advances it grows clouded, and when it is exhausted, it bears through it minute particles of water. This is because the latent heat of the steam has been partly converted into work, as its atomic has been changed into dynamic energy.

For each one horse power exerted 42.746114 units of heat must disappear per minute, since 33,000 units of work per minute make one horse power, and one unit of heat is the equivalent of 772 units of work. The unit of heat, or heat unit, is fully explained in another part of this work, but it will bear repetition here. The heat unit is the amount of heat required to raise one pound of water one degree of the Fahrenheit scale from a temperature of 32°. It has also been observed that the heat units increase at the rate of .305 for every degree of temperature above 212°.

TO DETERMINE THE NUMBER OF HEAT UNITS EXPENDED PER HORSE POWER.

RULE.—Divide the number of units of work (33,000) *in one horse power by the units of work* (772) *performed by one unit of heat.*

Then we have:

$$\frac{33000}{772}=42.746114\text{— Units of heat expended in one horse power exerted.}$$

At the pressure of 80 pounds per square inch by steam gauge, one pound of steam generated at that pressure requires an expenditure of 1212.6 units of heat to raise the water from zero to 80 pounds pressure and evaporate it at that pressure, and one pound of water at that pressure has required the expenditure of 326.6 units of heat to raise it to that pressure from zero. So that 886 units of heat are required to evaporate one pound of water from and at a pressure of 80 pounds per square inch. There must, therefore, be found by condensation in the cylinder, nearly 3 pounds of water per hour for each one horse power exerted.

$$\frac{42.746114\times 60}{886}=2.89476+\text{ Pounds of water condensed per horse power per hour.}$$

If this steam is expanded to and exhausted at the atmospheric pressure, .34 of a pound of water will be re-evaporated by the heat set free during the expansion, by the fall of the boiling point, which has fallen from 326.6 heat units to 212.9 heat units, showing a loss of 113.7 heat units.

At the atmospheric pressure one pound of water contains only 212.9 units of heat, and 965.7 units are required to evaporate it at that pressure.

Then we have:

$$\begin{array}{r l} 326.6 & \\ 212.9 & \\ \hline 113.7 & \text{Heat units expended.} \end{array}$$

Then, multiplying the heat units expended by the weight of water condensed per horse power per hour, and dividing the product by the heat units required to evaporate one pound of water at a temperature of 212° Fahrenheit, or 212.9 heat units, we have:

$$\frac{113.7 \times 2.895}{965.7} = .34+ \quad \text{Decimals of a pound of water re-evaporated.}$$

This leaves 2.555 pounds of water which must be exhausted in the form of water unaccounted for by the indicator.

If the expansion is carried down to 4 pounds absolute pressure then one pound of water would contain only 153.396 units of heat, but it would require 1007.229 units to evaporate it; so that .5 of a pound would be re-evaporated, leaving 2.395 pounds unaccounted for by the indicator at the termination of the expansion.

If this steam were exhausted at only 2 pounds pressure absolute, a slight further re-evaporation would take place, but there must still remain in the exhaust 2.33 pounds of water per horse power per hour. The exhaust must therefore always be wet.

Why is not this disappearance of the steam amounting, in ordinary good practice, to 10 per cent. of the whole quantity, shown by the indicator? The answer is twofold: First, that which takes place before the steam is cut off the indicator can not show; second, the steam which thus disappears during the expansion is restored—(1) by the re-evaporation of a small part of the water formed by the work done on the expansion itself; and, (2) by the partial re-evaporation of the water present in the cylinder at the point of cut off. This latter embraces the water formed by work done before the cut off, that which was formed by condensation of the entering steam, and that which the entering steam brought with it, either from the boiler or condensed in the pipes and chest. In the case above supposed of 80 pounds initial pressure, exhausting to the atmosphere, not one-twentieth of the water formed during the expansion is re-evaporated, because the average fall of the boiling point from the successive points in the expansion, is less than one-half its extreme fall; but of the water in the steam at the point of cut off, between one-eighth and one-ninth will reappear as steam, as the boiling point falls. (965.7÷113.7=8.5—.)

This re-evaporation, where there is no steam jacket or superheating, always restores the whole loss of pressure from work done on the expansion, and generally adds to it, sometimes considerably. In fact, it requires a good deal of additional heat to be imparted to the steam, in one or both the above ways, to bring the expansion curve below the isothermal line, or that theoretical curve that is obtained when the product of the volume into the pressure is a constant quantity.

Although the pressure at the end of the expansion thus always accounts for all the steam in the cylinder at the point of cut off, and often considerably more, still the water supplied to the boiler is not accounted for. The fact, always observed, that the higher the terminal pressure above the isothermal line, the smaller the portion of the water that is accounted for, is at least partly understood, when we remember that this excess is produced by re-evaporation of water in the cylinder, which, so far as the heat of vaporization is supplied from the water itself, by fall of the boiling point, will represent from eight to nine times as much water remaining unevaporated. In addition to this, however, an uncertain quantity of heat is imparted during the expansion by the hotter surfaces.

Having thus described this phenomenon of the conversion of heat into work in the cylinder, and explained some of the indications of the diagram, we will now consider the obtainable limit of economy.

ATTAINABLE LIMIT OF ECONOMY.

The heat that must be lost by conversion into work is a fixed quantity, namely, 2565 units per horse power per hour.

$$\frac{33000 \times 60}{772} = 2564.76683+$$

The practical problem is, how little heat can be put into the water and supply this quantity for conversion into work? At 100 pounds pressure each pound of steam contains 1217 units of heat, of which, for a condensing engine, about 1100 units need to be supplied in the boiler.

The above	number,	2565 is	.0933	of heat thus added	to	25	lbs.	of water.
"	"	2565 "	.1166	"	"	20	"	"
"	"	2565 "	.1555	"	"	15	"	"
"	"	2565 "	.2000	"	"	11.66	"	"

An economy better than the third one is attained by this engine. Theoretically, this economy is to be attained by twenty-fold expan-

sion, in two cylinders. Three-fourths of all the work is then done by the expansion of the steam, and, with a vacuum in the cylinder of 26 inches of mercury, the steam accounted for by the indicator will not exceed 10 pounds per horse power per hour.

But why not expand to the same extent in a single cylinder?

There are three important reasons. The principal one is as follows:

Heat is lost in cylinders by being imparted to their interior surfaces by the entering steam, which condenses on them, and during the exhaust is re-evaporated, taking up again the heat it had parted with, and conveying it to the atmosphere or the condenser. Water brought over with the steam does not add to this action; so also water formed in doing work passes out unevaporated, except by the fall of its own boiling point; since no more water can be evaporated from the surfaces than has been condensed upon them. But this alternate condensation and re-evaporation must always go on to the fullest extent possible under the conditions, which, with a metal of given conducting power, are the time, the surface, and the difference between the initial and the exhaust temperatures. The cylinder in this case, however, must be large and the surface extensive.

The difference of temperature between steam of 100 pounds gauge pressure and a vacuum of 26 inches, is $337°—126°=211°$, or 31° more than the difference between boiling water and ice. Under these conditions, of the small amount of heat entering the cylinder at the commencement of each stroke, by far the greater part would be lost; being locked up in the metal while the work was being done.

Steam jacketing and superheating are the means employed to prevent or diminish this action; but this case is quite beyond their power to remedy, unless the superheating is carried to an extent that would be destructive. On account of this loss of heat, it has been practically settled that, even with the use of the steam jacket and fairly superheated steam, six expansions are about as many as will give any gain in a single cylinder. But these give, theoretically, only 60 per cent. of the gain that is to be derived from a twenty-fold expansion.

The second reason why a twenty-fold expansion in a single cylinder is not of any advantage is, that the waste room is then about equal to the piston displacement, up to the point of cut off. It is obvious that, on this account also, six expansions are as many as can give any gain.

The third reason is, that such extreme variations of pressure, at the opposite ends of the stroke, are undesirable from a mechanical standpoint.

By the use of the two cylinders, according to the plan here pre-

sented, the steam is cut off in each one at from one-fourth to one-fifth of the stroke, and we gain in the following ways:

In the first, or high pressure cylinder, in round numbers, the fall of temperature is one-half as great, and the extent of surface is one-quarter as great, giving only one-eighth the amount of alternate condensation and re-evaporation. Now this amount can be almost entirely prevented by steam jacketing and superheating. If no water has been condensed on these surfaces they can be cooled during the exhaust only by evaporation of the small portion of the water from work done diffused through the steam that comes in contact with them, and the heat thus lost may be restored by conduction from the jacket, and by loss of the superheat, without causing any condensation of the entering steam. If any heat should be carried away it is not carried direct to the condenser, but is utilized in the second cylinder.

Again, the waste room bears but one-fourth or one-fifth the proportion to the piston displacement up to the point of cut off that it did in the single cylinder, and the compression, commencing at the same pressure at which the expansion terminates, or higher, is carried up to the initial steam pressure, or nearly so, thus avoiding loss from the waste room in this cylinder almost entirely. So much for the first cylinder.

We now come to the point where compounding frequently fails. In theory the work done in the second cylinder is but a continuation of the expansion, as if the first cylinder were five times as long, and the expansion, instead of stopping at 25 pounds, would continue down to 5 pounds pressure. This being impractical, both for mechanical and refrigerative reasons, the problem is to transfer the steam, without loss, to another cylinder four times as large, in which the expansion can be completed. But here is the difficulty. Usually a large part of the steam disappears in the transfer. This disappearance is caused partly by the filling of passages and clearance, but chiefly by condensation of the steam as it enters the low pressure cylinder. There being no boiler to furnish an unlimited supply, but the supply being only the quantity discharged from the first cylinder, the loss from condensation of the entering steam becomes plainly apparent.

The fall of pressure is hardly ever less than 8 pounds, while it frequently reaches 25 pounds in some engines, and consequently a corresponding loss of heat. How to prevent this loss in the second cylinder without superheating has been the question. The exhaust from the high pressure cylinder is already charged with water from the work done, and which must escape mostly unevaporated. If the steam in this state must go into the low pressure cylinder, there can be no hope of first-rate economy.

STEAM JACKETED INTERMEDIATE RECEIVER.

Mechanical ingenuity has devised a number of contrivances to prevent any undue loss of heat in the steam in passing from the first to the second cylinder; an intermediate receiver, provided with a steam jacket, is employed. The pressure maintained in the receiver is that to which the steam is expanded in the high pressure cylinder, or above this; there is no fall of pressure suffered in exhausting from this cylinder.

The low pressure cylinder takes its steam from the receiver as from a boiler, and cuts it off and works it expansively, as a separate engine would do. This action is shown in the following diagrams, taken from one of these compound engines, 12 inch and 21 inch by 24 inch stroke, at 180 revolutions per minute.

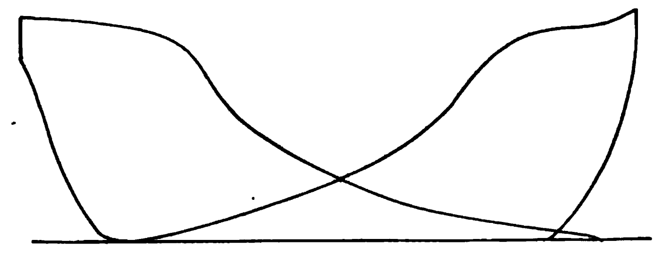


Fig. 446.

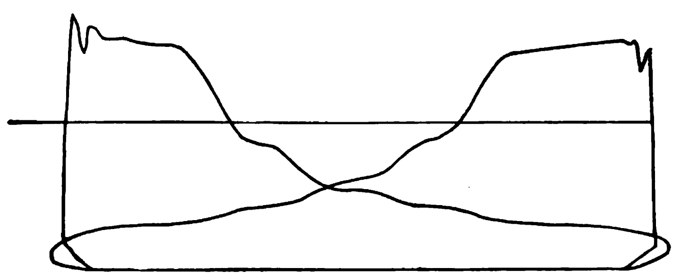


Fig. 447.

The steam exhausted from the high pressure cylinder is wholly accounted for in the low pressure cylinder. There is a slight fall of pressure sufficient to produce the current through the ports, but this only requires the cut off to take place a little later. The product of the volume into the density is everywhere the same, giving the same weight of steam from the cut off in the high pressure cylinder to the exhaust in the low pressure cylinder. To effect this the receiver is formed to serve first as a separator; part of the water contained in the

steam is caught and drawn off by a trap. This is let run to waste, being greasy and little of it.

The steam thus partly freed from water is dried and sufficiently superheated by the jacket, so that it enters the second cylinder under very nearly the same advantages that it had at first.

Both cylinders are steam jacketed, body and heads, and the special advantage in applying the jacket to the low pressure cylinder, is in the greater difference of temperature, combined with the more extended surfaces, which increases its efficiency to a very great extent. The object here is, by means of superheat retained, and by radiant heat from the walls, to supply, as it is required, the heat to be converted into work, so as to obtain in this cylinder, and to maintain to the end of the stroke, all the pressure possible from the steam supplied by the first cylinder, without re-evaporation except from fall of the boiling point. This cylinder is kept dry and the temperature low, therefore, by proper lubrication, no injury can result to it.

There are no passages between the cylinders to be filled anew at each stroke, the receiver extends from valve to valve, and only about 5 per cent. waste room has to be suffered. The water condensed in the jackets is returned to the boilers by a separate pump placed below the receiver.

The diagrams shown in Figs. 446 and 447, were taken from one of these compound engines whose cylinders give a ratio of 1 to 3 by 24 inch stroke. The better ratio between the cylinders, especially for larger engines, would be 1 to 4. In this engine the full indicated horse power was to be 200, with 100 pounds initial pressure; the work, however, was not all on, the average load being 105 horse power. These diagrams were among the largest taken, and represent 115 horse power.

The cut off in the high pressure cylinder was impared by a long pipe, with several elbows; that in the low pressure cylinder is given by corresponding port openings. The waste room in clearance and port is excessive, being 6.5 per cent. in the high pressure cylinder, and 8.33 per cent. in the low pressure cylinder. The steam was expanded about sixteen times, and the same weight of steam was accounted for on the diagrams, from the cut off in the first cylinder to the final release in the second. The water drained from the receiver was 50 pounds per hour, and that from the jackets 210 pounds per hour, making together about $2\frac{1}{2}$ pounds per horse power per hour. This was thrown away, as the means of returning it to the boilers were not then provided. The steam was not superheated, and in fact was not dry, as no means for moderate superheating had at that time been found, the want of which seriously affected the result.

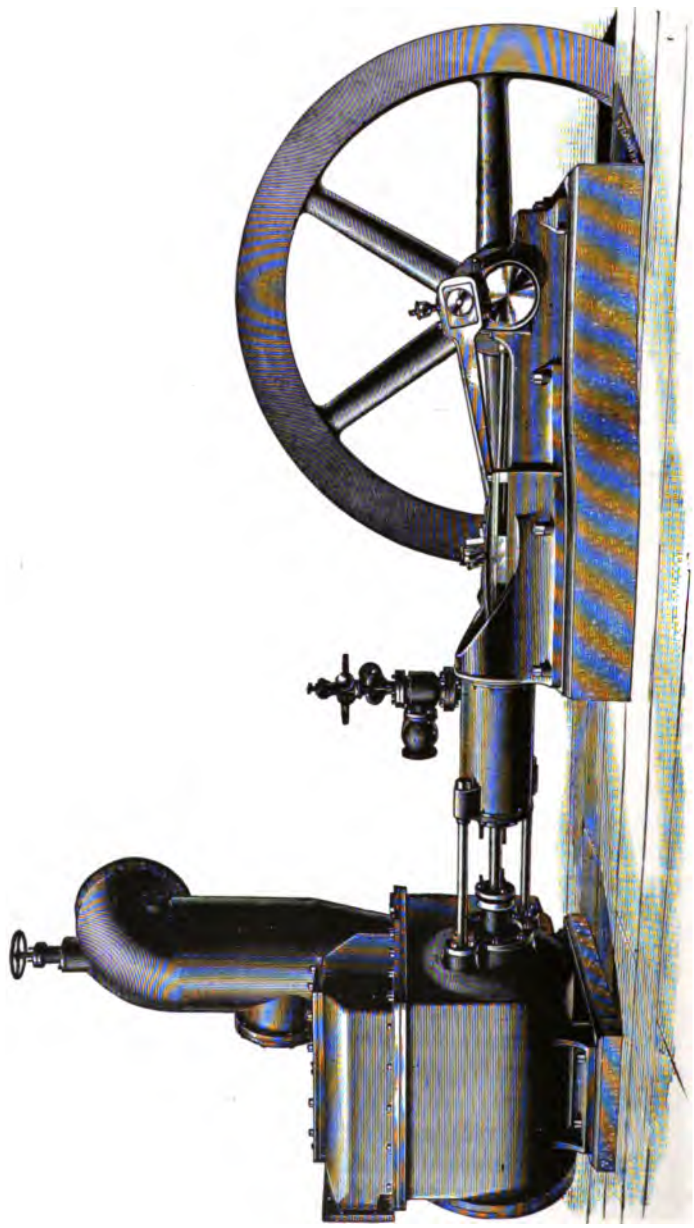


Fig. 448.

The steam accounted for by the indicator averaged 11.75 pounds per horse power per hour, and the quantity drained, as before described, brought this up to 14.25 pounds. The quantity of water pumped into the boiler was 18.5 pounds, showing 4.25 pounds still unaccounted for. This result shows just what the system is capable of and what was required to perfect it.

INDEPENDENT AIR PUMP AND CONDENSER.

Fig. 448 is a perspective view of independent air pump and condenser, used in connection with these engines.

INDICATOR DIAGRAMS.

The following cards have been taken from various sizes of these engines and under different conditions:

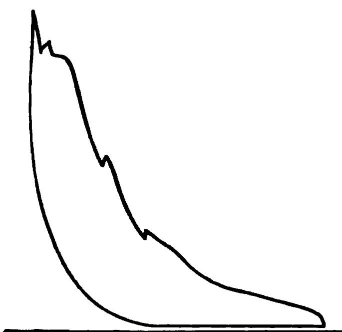


Fig. 449.

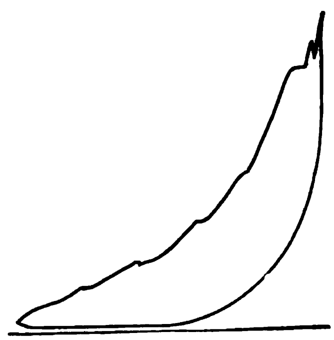


Fig. 450.

Steam pressure, 80 pounds.
Diameter of cylinder, 6 inches.
Stroke, 12 inches.
Revolutions per minute, 305.
Piston speed, 710 feet.
Horse power, front end, 6.28.
Horse power, back end, 6.45.
Total horse power, 12.73
Water consumption per horse power per hour, by card, 22.13 pounds.

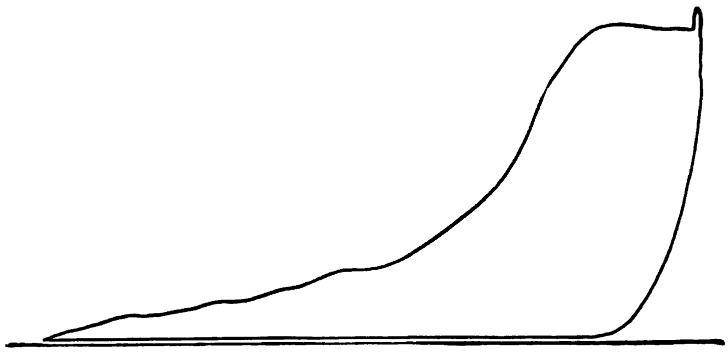


Fig. 451.

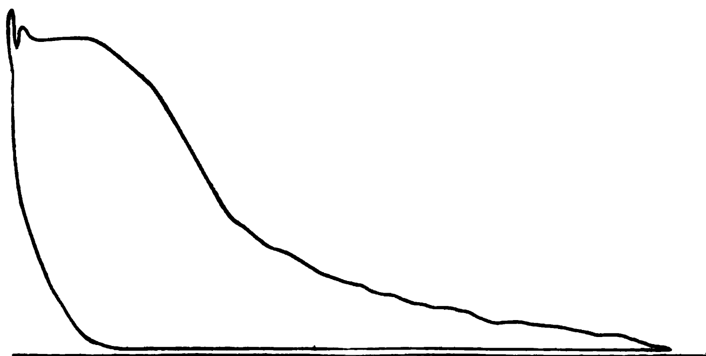


Fig. 452.

Steam pressure, 55 pounds.
Diameter of cylinder, 12 inches.
Stroke, 20 inches.
Revolutions per minute, 230.
Piston speed, 766 feet.
Horse power, front end, 24.15.
Horse power, back end, 23.65.
Total horse power, 47.80.
Water consumption per horse power per hour, by card, 22.10 pounds.

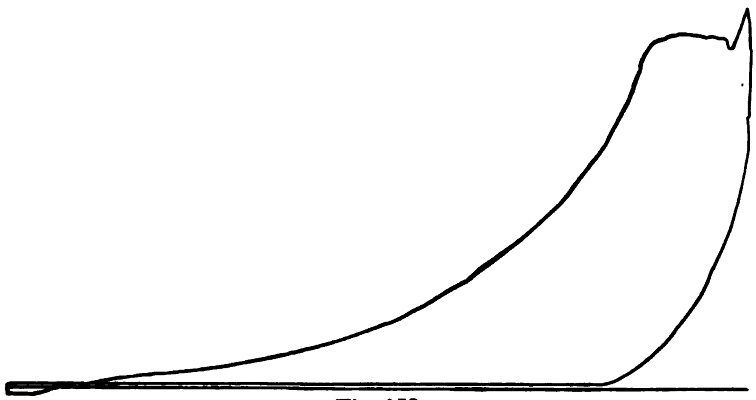


Fig. 453.

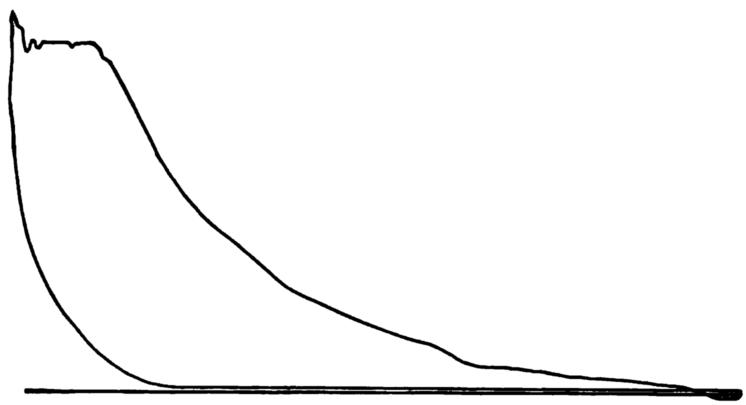


Fig. 454.

Steam pressure, 68 pounds.
Diameter of cylinder, 16 inches.
Stroke, 30 inches.
Revolutions per minute, 125.
Piston speed, 625 feet.
Horse power, front end, 36.48.
Horse power, back end, 37.05.
Total horse power, 73.53.
Water consumption per horse power per hour, by card, 18.72 pounds.

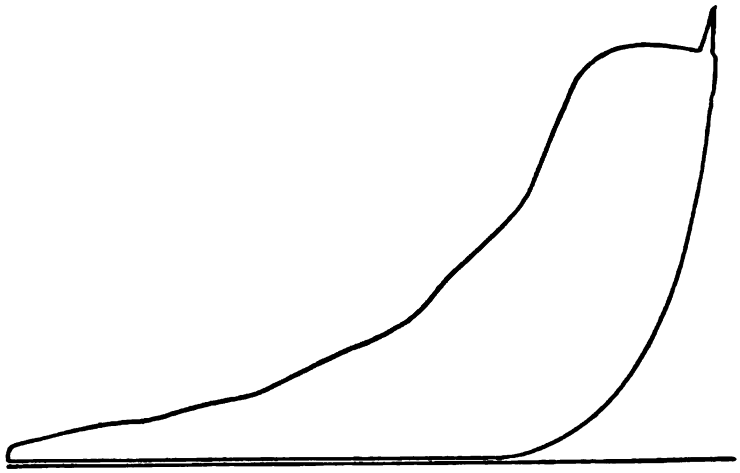


Fig. 455.

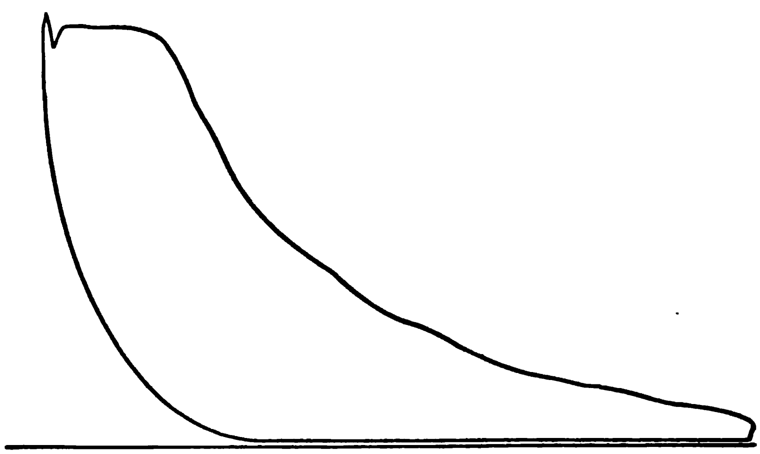


Fig. 456.

Steam pressure, 63 pounds.
Diameter of cylinder, 54 inches.
Stroke, 66 inches.
Revolutions per minute, 82.
Piston speed, 902 feet.
Horse power, front end, 695.1.
Horse power, back end. 687.3.
Total horse power, 1382.4.
Water consumption per horse power per hour, by card. 19.26 pounds.

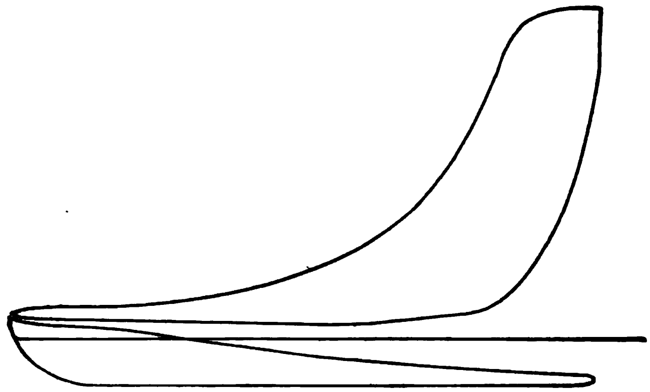


Fig. 457.

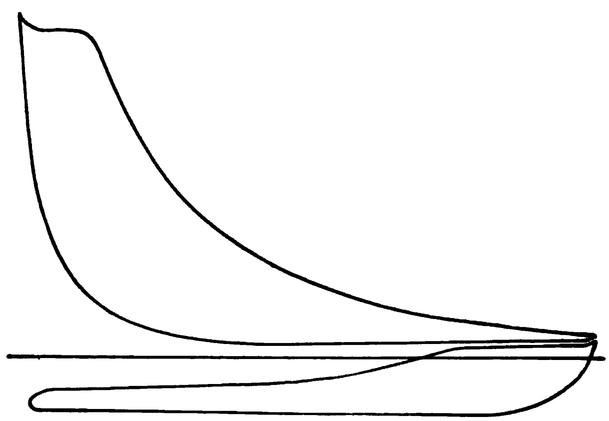


Fig. 458.

TANDEM COMPOUND ENGINE.

Initial steam pressure, 90 pounds.
Diameter of high pressure cylinder, 12 inches.
Diameter of low pressure cylinder, 22 inches.
Stroke, 20 inches.
Revolutions per minute, 160.
Piston speed, 533 feet.
Horse power high pressure cylinder, 45.5.
Horse power low pressure cylinder, 56.73.
Total horse power, 102.23.
Water consumption per horse power per hour, by card, 14.85 pounds.

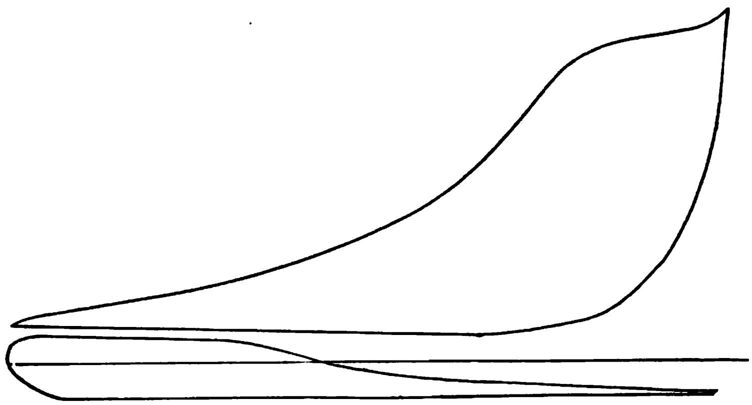


Fig. 459.

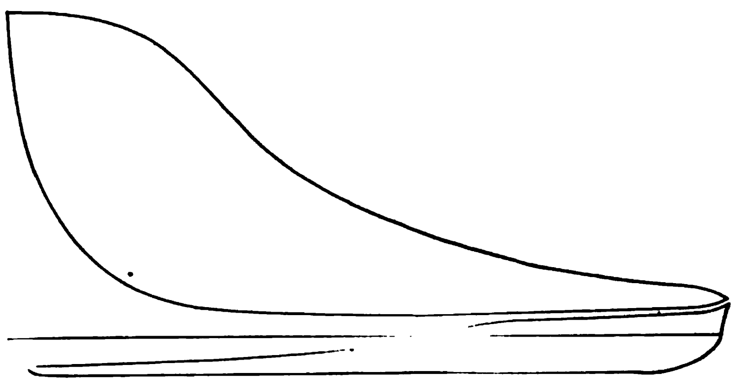


Fig. 460.

Initial steam pressure, 120 pounds.
Diameter of high pressure cylinder, 24 inches.
Diameter of low pressure cylinder, 46 inches.
Stroke, 42 inches.
Revolutions per minute, 116.
Piston speed, 812 feet.
Horse power high pressure cylinder, 444.2.
Horse power low pressure cylinder, 424.1.
Total horse power, 868.3.
Water consumption per horse power per hour, by card, 12.87 pounds.

TABLE OF PORTER-ALLEN HIGH PRESSURE ENGINES.

Dimensions of Cylinder.		Speed.		Horse Power Constant for One Pound M. E. P. One Revolution.	Capacity in Horse Power. 85 Pounds Initial Pressure.			
Bore in Inches.	Stroke of Piston.	Revolutions per Minute.	Piston Speed in Feet per Minute.		Rated Horse Pow'r at approximately ¼ Cut Off.	Economical Range, $\frac{1}{5}$ to $\frac{1}{3}$ Cut Off.		Maximum Horse Power ½ Cut Off.
8	16	280	746	.0040	45	39	56	72
10	12	300	600	.0044	53	44	66	84
9	16	280	746	.0051	60	50	71	92
11	12	300	600	.0057	64	52	80	102
10	20	230	766	.0079	75	63	90	108
12	12	300	600	.0069	80	68	95	122
12	14	290	677	.00788	91	74	114	148
12	20	230	766	.0114	107	91	131	170
13	14	290	677	.00927	107	87	134	174
14	14	290	677	.01077	124	101	156	202
13	24	200	800	.0166	130	112	160	208
14	16	275	733	.01226	134	109	168	218
15	16	275	733	.01410	154	125	194	251
14½	24	200	800	.0200	160	140	200	260
16	16	275	733	.01607	175	142	221	286
16	18	255	765	.0182	180	145	225	290
16	20	230	767	.0200	180	145	225	290
17	18	255	765	.0206	200	175	250	325
16	30	165	825	.0304	200	175	250	325
18	18	255	765	.0231	225	185	272	348
18	20	230	766	.0257	235	206	295	384
18	30	165	825	.0385	250	222	317	412
20	20	230	767	.0314	285	230	360	455
20	36	140	840	.0571	320	279	399	519
22	24	200	800	.0460	350	295	450	575
22	36	140	840	.0690	400	338	483	627
24	24	200	800	.0545	410	345	509	652
24	42	125	875	.0958	480	419	598	778
26	42	125	875	.1126	560	492	703	914
28	42	125	875	.1304	645	570	815	1,059
28	48	112½	900	.1490	670	586	838	1,089
30	42	125	875	.1482	740	656	910	1,289
32	48	112½	900	.1949	870	767	1,096	1,425
36	48	112½	900	.2467	1,100	971	1,387	1,803
40	48	112½	900	.3046	1,360	1,199	1,713	2,227
44	48	112½	900	.3686	1,600	1,451	2,073	2,695
44	66	82	902	.5066	1,650	1,493	2,146	2,750
48	66	82	902	.6030	2,000	1,730	2,472	3,213
48	72	75	900	.6580	2,000	1,730	2,472	3,213
54	66	82	902	.7634	2,500	2,180	3,072	4,000

CHAPTER XXIV.

THE WEIR-HARDEN COMPOUND ENGINE.

This is a recent invention, and bids fair to become a most formidable competitor for first place among the leading American steam engines. It is a separate and distinct type, and presents many new and novel features worthy of careful study on the part of engineers and students of steam engineering. It is therefore presented in detail, with full instructions in regard to construction and arrangement of valve gear, shaft, governor, method of adjusting and regulating the same, and setting and adjusting the valves and valve gear.

This engine is provided with two valves which regulate and control the flow of steam to and from the cylinders, as shown in Figs. 461, 465 and 466.

Fig. 461 is a front elevation of vertical engine partly in section.

Fig. 462 is an end elevation of vertical engine.

Fig. 463 is the rear pillow block.

Fig. 464 is a cross section through the cylinder and steam chest.

Figs. 465 is a longitudinal section of steam and exhaust chambers of steam chest, showing the arrangement of the valves, steam passages, balancing and other features of the valves.

Fig. 466 is a longitudinal section through the center of the cylinder, piston, valves and steam chest.

A (Fig. 465) is the cut-off valve; B the main valve; C the relief valve; D D D D, openings in the side of the steam chest for the admission of live steam; 1, balance recess of cut-off valve; 2 2, steam passages through cut off valve connecting auxiliary recesses 5 5, with balance recess 1; 3 3, openings into cut-off valve for main steam supply; 4, main admission port from supply openings 3 3 for admission of steam alternately into ports 9 9, in center partition plate A′, which plate separates the live steam part of the chest from the exhaust steam part of the chest, the whole being cast in one casting; 5 5, auxiliary recesses in the under side of the cut-off valve; these communicate alternately with the outer edges of the valve seat of partition A′ and recess 10, and admit steam to ports 9 9 from the live steam chest, and to ports 9 9 from port 4, the whole forming a multiple steam opening equal to four times the amount of valve movement, so that when

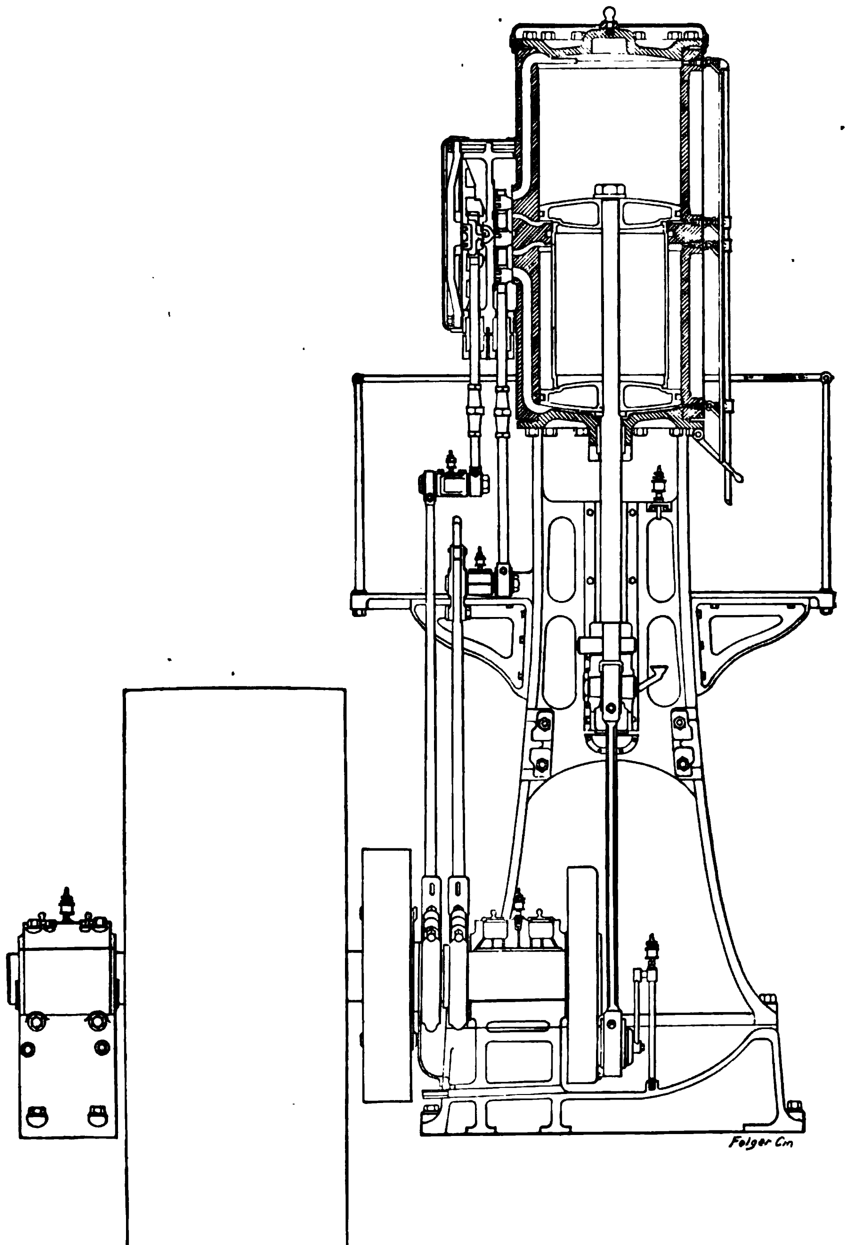


Fig. 461.

FRONT ELEVATION, PARTLY IN SECTION.

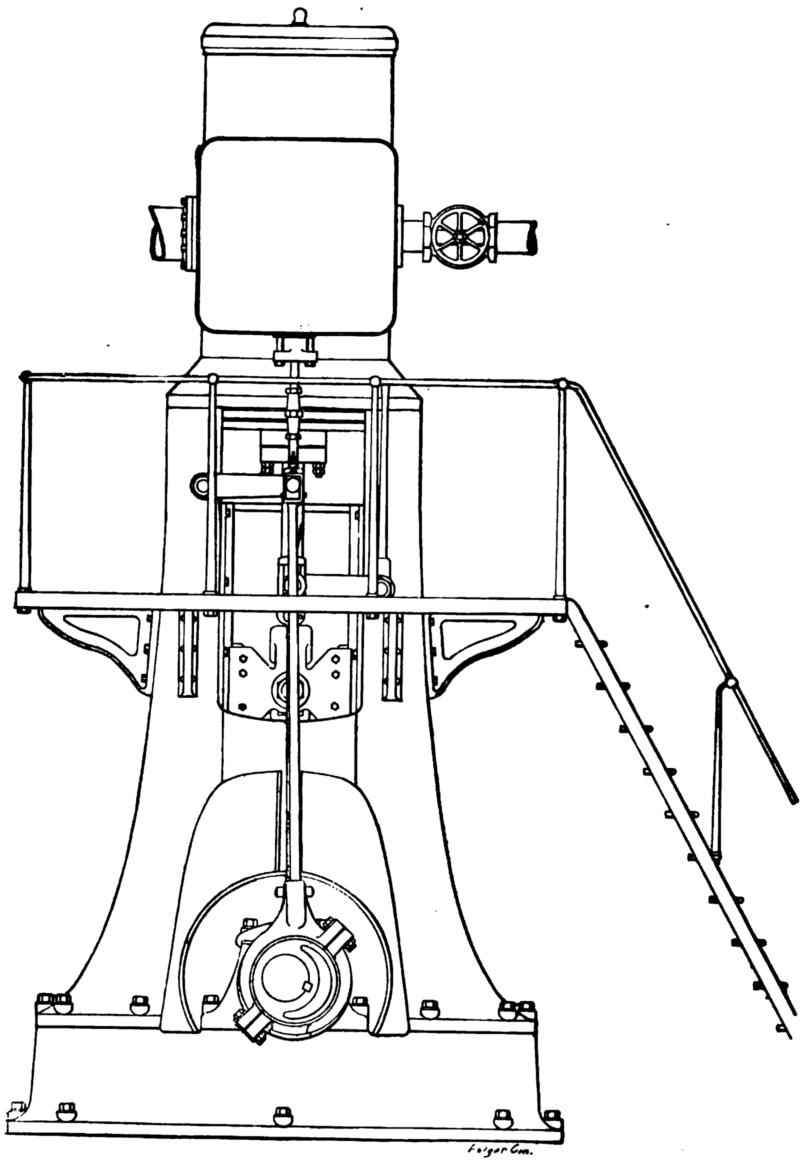


Fig. 462.

END ELEVATION.

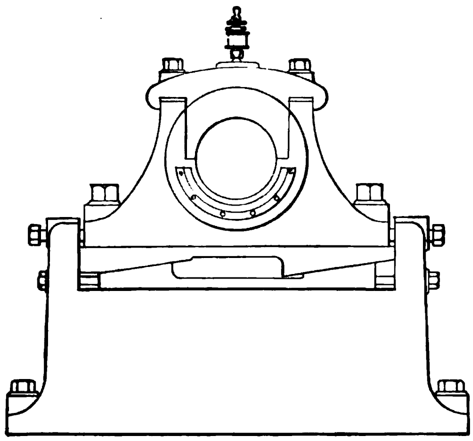


Fig. 463.

END ELEVATION OF PILLOW BLOCK.

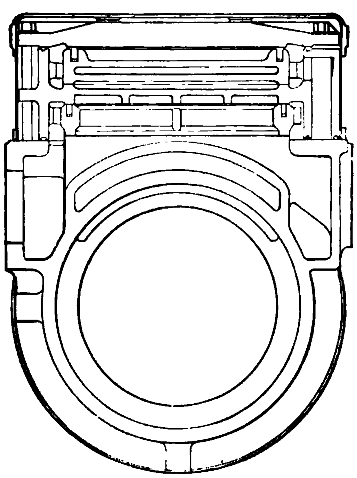


Fig. 464.

CROSS SECTION THROUGH CYLINDER AND STEAM CHEST.

the valve has traveled one-eighth inch after commencement of opening the combined opening will be equal to one-half inch; 6 6, cross packing strips in the upper surface of the cut-off valve; there are also two side strips, not shown in the engraving, the whole forming a steam tight enclosure of proper area for balancing the valve; 7, a plate cast on the under side of the steam chest cover, which forms a surface for the movement of the packing strips; 8, space cored between 7 and the steam chest cover for the reception of lubricants, which, for locomotives, are generally admitted through the steam chest cover; 9 9, main ports leading from the cut-off valve through partition plate A′ forming port 11; 10, auxiliary recess in the seat of the steam chest partition, communicating alternately with recesses 5 5 for the transmission of steam to ports 9 9; 11, port formed at the junction of 9 9 for the admission of steam into the main valve; 12 12 12 12, ports through the main valve, connecting with spaces 13 13 13 13, forming passageway for the steam to ports 16 16 from port 11; lower spaces 13 13 form a passageway for the steam from port 11 into the high pressure part of the cylinder through ports 16 16 alternately, and from port 16 of the high pressure to port 17 of the low pressure part of the cylinder—the upper spaces 13 13 are employed for counterbalancing the pressure in lower 13 13, and for conveying steam into ports 12 12; 14, a small passageway through the center bridge of the main valve B to recess 18, to counterbalance the pressure from port 11; 15 15 15 15, cross packing strips in the upper side of the main valve—there are also two longitudinal strips, one on each side of the valve, forming a bearing against the end of the cross strips, the whole enclosing an area sufficient to balance the valve—the side strips are not shown in the illustration; 16 16, ports leading into the high pressure part of the cylinder; these ports answer the double purpose of a passageway for the high pressure steam into the high pressure part of the cylinder and exhausting it after expansion into the low pressure part of the cylinder through recesses 13 13 and ports 17 17; 17 17, receive the exhaust steam from high pressure ports 16 16 for low pressure expansion, after which they serve as passageways for the exhaust steam from the low pressure ends of the cylinder to the exhaust steam chest 19 19; 18, recess in valve seat for the main valve, for counterbalancing pressure on top of main valve from port 11; 19 19, exhaust steam chest, this has two openings, one at each end of the cylinder, leading into a longitudinal passageway to an outlet into a heater or to the atmosphere; or, as in the case of a locomotive, through the saddle to the exhaust nozzle; *a* the valve stem yoke of the cut-off valve; *a′* the valve stem of the cut-off valve; *b* the valve stem yoke of the main valve; *b′* the valve stem of the main valve.

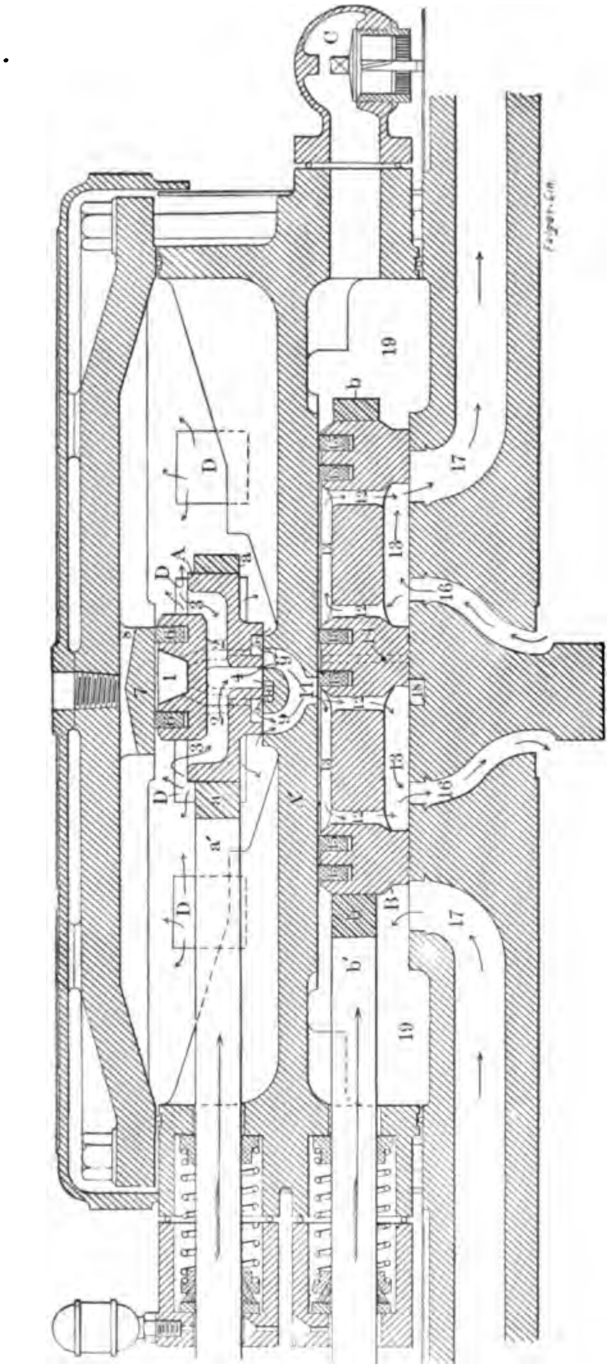


Fig. 465
LONGITUDINAL SECTION THROUGH CENTER OF STEAM CHEST.

E (Fig. 466) the cylinder, is a continuous casting; *d d*, partition ring cast in the center of its bore. This ring is of such depth as to give the differential area between the high and the low pressure piston surfaces, which surfaces being made in a ratio of about 3 to 1; *e*, steam packing ring in partition ring *d*, forms a bearing on the outer surface of the shell *f* of the piston F. This piston is made in three parts, two followers *g g*, and a cylindrical shell *f* interposed between them. It will therefore be seen that there are four piston faces in one cylinder, two annular surfaces *h h* of a smaller area for high pressure steam, and two *h' h'* of a larger area for low pressure steam. There are two steam packing rings *i i*, one in each follower. All of the steam packing rings are of the Harris type, made in sections, with a spiral spring at each joint, set into a socket just large enough to hold and guide it. In the partition ring *e*, the springs are placed in a reverse position from those in the followers, so as to close the ring around the barrel, or cylindrical shell, of the piston. The piston rod *j* passes through both followers and the cylindrical shell, and holds all of the parts of the piston firmly together by means of a steel collar and shoulder against one of the followers, and a nut on the other to tighten up against it.

The ports are designed to allow a large flow of steam, and being long and narrow, give a quick opening and equally quick closure. The high pressure ports are very short from the cut-off valve to the cylinder, and much longer than the diameter of the high pressure part of the cylinder. The low pressure ports have an easy bend and are proportioned to suit the area of the low pressure part of the cylinder. The clearance in the high pressure ports is larger during expansion than during compression. This holds up the terminal pressure and gives a higher initial pressure for the low pressure part of the cylinder, while the clearance during compression is small; therefore, the waste of steam in filling the ports is materially reduced, and the necessity for early compression is, to a great extent, obviated. The steam is expanded directly from the high pressure side of the piston to the opposite or low pressure end of the piston; consequently, the fall in pressure from the terminal of the high to the initial of the low is very small without compressing the low-pressure steam.

THE MAIN VALVE.

The main valve is a plain skeleton slide valve, and is perfectly balanced by means of its own steam pressure. It gives the high pressure port a large opening before the piston starts on its return stroke, and when the piston has traveled a short distance the port is wide open, and remains so until the piston has traveled from one-half to five-eighths of its stroke. The compression can be varied without affecting

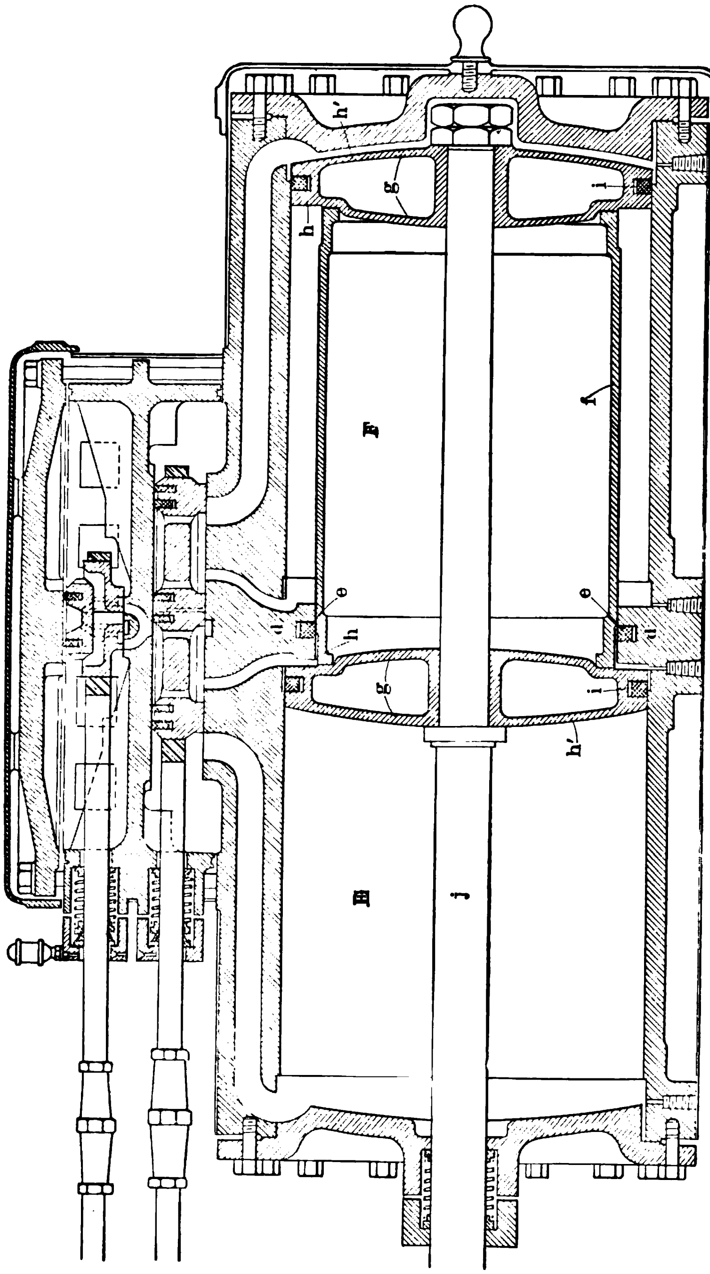


Fig. 466.
LONGITUDINAL SECTION THROUGH CENTER OF CYLINDER.

the cut off, and the travel of the valve is less than that in general practice. The exhaust ports from the low pressure end of the cylinder are open one-half the width of port when the engine is on its center, hence, giving a free exhaust of steam on account of the size of the ports.

This valve performs all the functions for controlling the admission of steam to the high pressure part of the piston, its passage from the high to the low pressure part of the piston, and the exhaust from the low pressure ends of the piston to the exhaust steam chest.

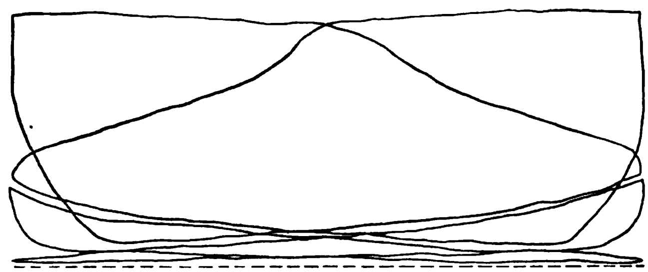
BALANCING FEATURE OF THE MAIN VALVE.

The main valve B is balanced by means of the ports 12 12 12 12, spaces 13 13 13 13, recess 18, and passage 14 through the center of the valve, as shown in Fig. 465. When the valve is in the center of its travel, and the central bridge closes port 11, and subjects the central portion of the valve to pressure from above, steam has access to recess 18 in the valve seat below through passage 14, and counterbalances from below the pressure on top of the valve, as the area of the recess is nearly equal to that enclosed by packing strips 15 15 in the central bridge of the valve. When port 11 is open and the steam has access to port 16 on that side of the valve, the valve is balanced on that side by the steam pressure being equal in spaces 13 13 on the same side. The ports 16 16 are always wide open, so that when the valve is admitting steam to the cylinder on one side, the expansion pressure of steam from the cylinder on the opposite side has access to upper space 13, through passage 12 12, and equalizes the pressure in lower space 13, and thus balances the valve on that side. The area on top of the valve subjected to steam pressure is just enough in excess of that at the lower face subjected to pressure to insure the valve being kept down on its seat just hard enough to keep it tight and free from leakage. The steam exhausting from the large or low pressure part of the cylinder has very little effect on the valves, as it exhausts into the exhaust steam chest at the outer edges of the valves.

THE CUT-OFF VALVE.

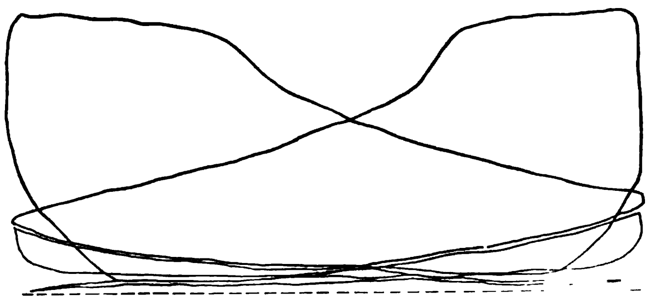
The cut-off valve can be operated to cut off from the initial line to any point up to $\frac{8}{10}$ of the engine stroke without any wire drawing of steam, or dropping of the steam line at high piston speed, as shown in diagrams 467, 468, 469, and 470, taken from this engine:

Diameter of high pressure cylinder, 10 inches.
Diameter of low pressure cylinder, 20 inches.
Ratio of cylinders, 1 to 4
Stroke, 20 inches.
Scale, 60.
Engine exhausting through heater.



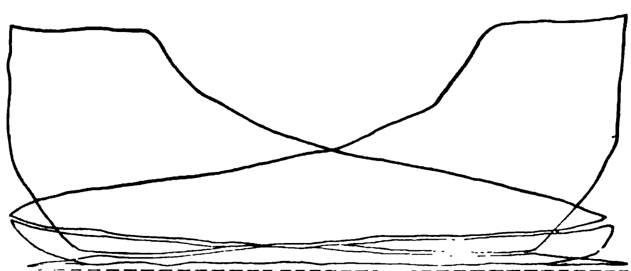
Boiler pressure, 83 lbs. Revolutions, 204. Speed, 680 feet per minute. $\frac{9}{16}$ cut off.

Fig. 467.



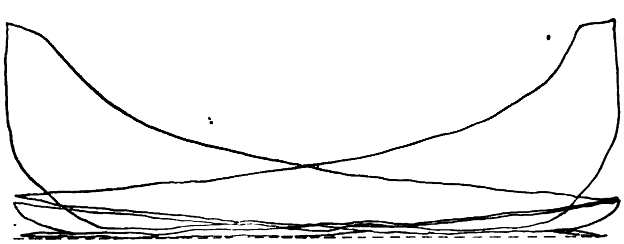
Boiler pressure, 90 lbs. Revolutions, 208. Speed, 693⅓ feet per minute. $\frac{5}{16}$ cut off.

Fig. 468.



Boiler pressure, 77 lbs. Revolutions, 208. Speed, 693⅓ feet per minute. ¼ cut off.

Fig. 469.



Boiler pressure, 70 lbs. Revolutions, 206. Speed, 686⅔ feet per minute. $\frac{1}{16}$ cut off.

Fig. 470.

The steam line is carried very nearly parallel to the atmospheric line to the point of cut off, which is sharp and well defined.

The lead of the valve commences when the crank is very nearly on the center, and when it is on the center the valve has very nearly a full port opening, which opening is obtained by the number of ports and the opening of the valve on each side of the ports simultaneously and in uniform proportion, so that the valve has nearly a full port opening when the piston starts on its return stroke, and does not contract the opening until the point of cut off is nearly reached, when the valve closes quickly, just as it opened quickly. As the auxiliary port in front of each main port closes, the port behind it opens in the same proportion, until a point is reached where both forward and backward openings are equal to each other, and also equal to the full port opening, when they are decreased until the ports are closed, which is done in a very short movement of the valve. It will, therefore, be seen that the operation described produces a sudden opening, an almost constant full port opening during the movement of the valve, and a sudden closure. The steam flows through the center of the valve from the upper side of both ends down through the main ports and under each end directly, and also through auxiliary passages, one-half going through the ports in the valve and the other half going in at and also under the ends of the valve through recesses into ports in the partition plate, making a multiple action, as shown in the following illustrations:

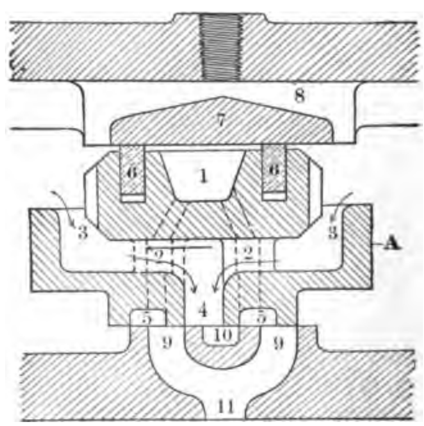


Fig. 471.

Fig. 471 shows the valve A. traveling in the direction shown by the arrow, and just at the point of opening.

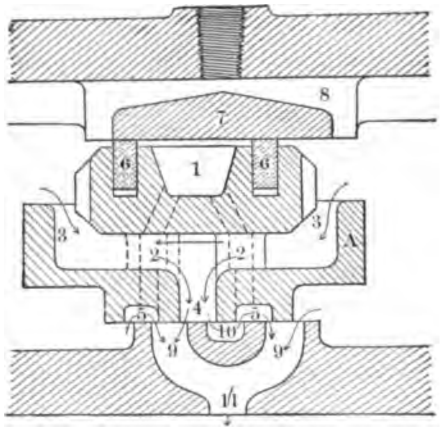


Fig. 472.

Fig. 472 shows the valve traveling in the direction shown by the arrow, and steam ports full opening. The ports open on one side as fast as they close on the other, and as the area of the ports at the valve face is twice as great as at the port entrance, the valve is required to travel only from the position shown in Fig. 471 to that shown in Fig. 472 to give a full port opening.

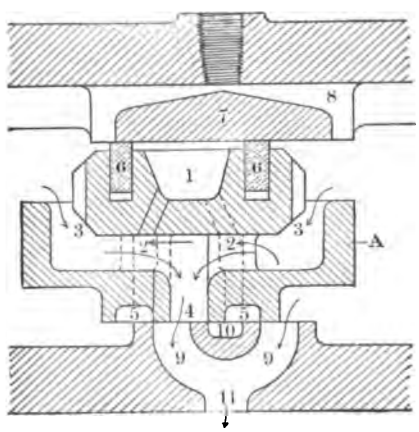


Fig. 473.

Fig. 473 shows the valve in its extreme position, in the direction shown by the arrow, and is full port open.

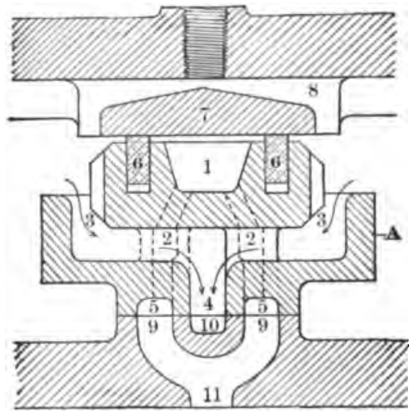


Fig. 474.

Fig. 474 shows the valve in the center of its travel, and ports closed.

BALANCING FEATURE OF THE CUT-OFF VALVE.

This valve A is so constructed as to be perfectly balanced during every part of its travel. The upper surface of the cut-off valve, as shown in Fig. 474, is provided with a recess 1, so that when the valve is subjected to steam pressure under it, the steam has access to recess 1 through passages 2 2, and the area enclosed by packing strips 6 6 is equal to the area of the lower face of the valve subjected to pressure; this balancing is performed when the valve is open and the steam pressure gets under it. When the valve is closed, and no pressure is operating on the lower surface, there is no pressure on the top of the valve on the space enclosed by the packing strips 6 6. If any pressure should come under the valve, whether from the cylinder or from leakage, it will find its way to the top of the valve through passages 2 2, and thus counterbalance the pressure on the valve below. It will therefore be seen that no matter what position the valve may be in, it will always be perfectly balanced.

THE GOVERNOR.

The cut-off valve is controlled by a shaft governor of peculiar construction, a description of which will be here given.

Fig. 477 is a plan view of the governor, showing the position of the parts at approximately full stroke.

Fig. 478 is a similar view, showing the position of the parts at the short or reduced stroke.

Fig. 479 is a section on line *x x* (Fig. 477).

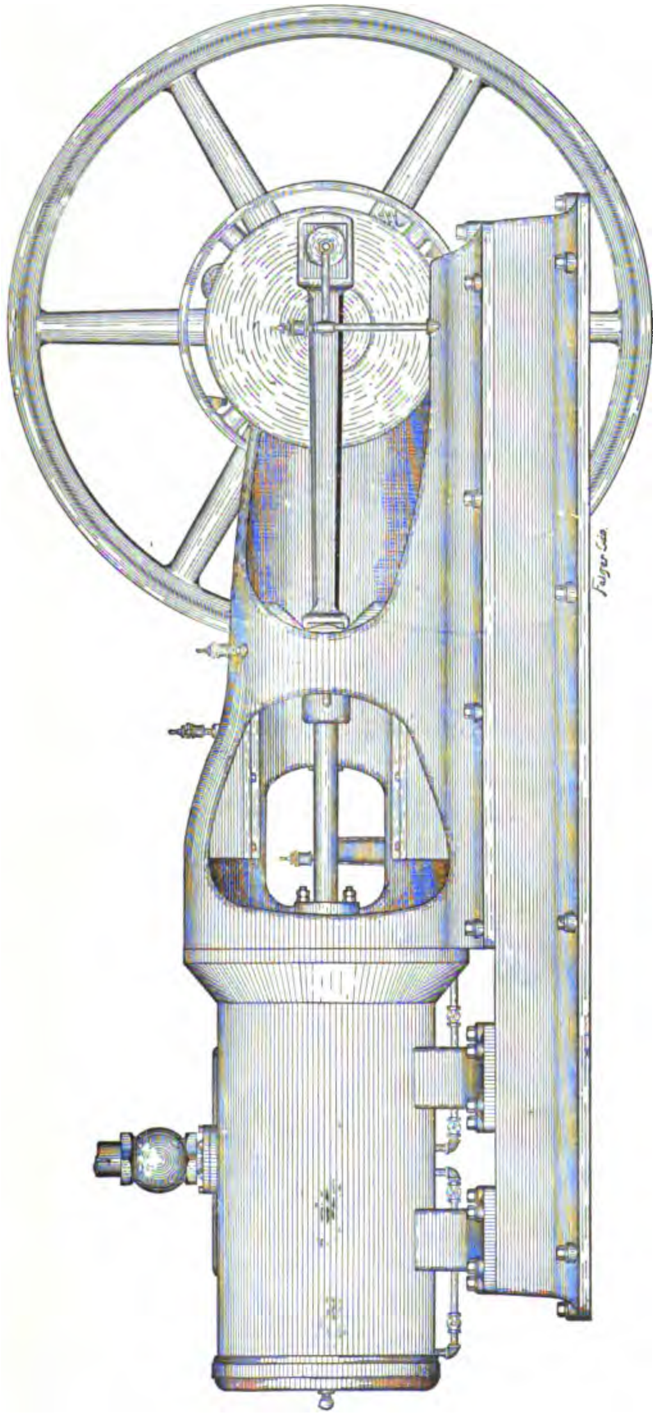


Fig. 475.
SIDE ELEVATION.

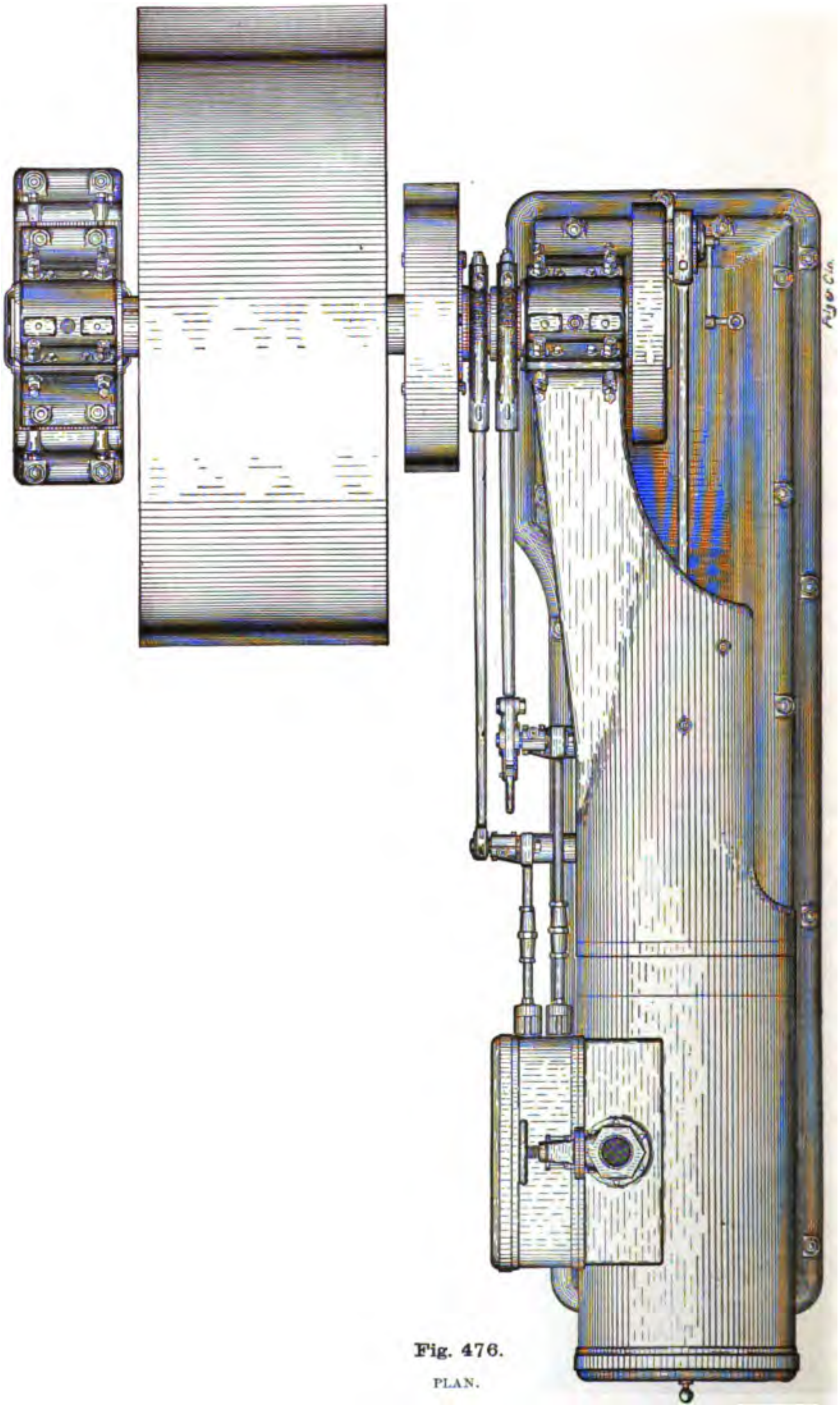


Fig. 476.

PLAN.

Fig. 480 is a plan view of the oscillating disc.

Fig. 481 is a cross section of the same.

Fig. 482 is a plan view of the eccentric ring.

Fig. 483 is a plan view of the pendulous lever.

Fig. 484 is a modification of Fig. 477.

Fig. 485 is a vertical section on line *x x* (Fig. 484).

Fig. 486 is a sectional elevation on line *y y* (Fig. 484).

In Fig. 479, 1 represents the hub of the governor wheel; 2 the rim; 3 the main shaft of the engine; 4 the arms connecting the hub and rim together.

In Fig. 483, A represents the pendulous lever which supports the eccentric ring shown in Fig. 480. The pendulum is provided with an oblong opening 5, through which passes the main shaft 3. 6 represents the center on which the pendulum is hung, and it is secured to boss 7 of one of the arms by the axial bolt passing through lug *h* and boss 7, as shown in Fig. 479.

In Fig. 482, B represents the eccentric ring. It is grooved around its periphery, in which groove the eccentric strap is journaled. It is provided with slotted openings 8 8. 9 represents bosses on the pendulum A, upon which the eccentric rests and is secured by means of bolts passing through the oblong openings of the eccentrics and tapping into the bosses 9 (Fig. 483). This eccentric can therefore be adjusted radially upon the lever A by loosening the bolts and moving it on the bosses 9, to lengthen or shorten the valve travel by increasing or decreasing the throw of the eccentric.

In Fig. 480, C represents the eccentric-moving lever plate. Its movement is rotary and journaled upon the shaft 3 (Fig. 479). It is provided with cam or eccentric guides D D′. When the device is to be used for a right-hand governor, only one such guide is employed. If it is desired to be used either right or left, a secondary cam or guide D′ is provided.

In Figs. 477 and 478, E represents a slide. It may be either of the pin or other suitable form to be held in the guide D, in which it journals; *a* represents an ear to which the pendulous lever A is bolted. The opposite ear on the lever is employed only when the governor is required to run in the reverse direction. F represents centrifugal or weighted arms counterpoised by springs I I. Two arms are used in the preferred form of construction, but only one may be employed, as shown in Fig. 484. These arms are each journaled upon a center, and one end of the arms is hinged to the eccentric-moving plate, and the other arm is connected to the spring which opposes the action of the weighted arms. When the speed of the governor wheel is increased, the weighted arms move outwardly and compress the spring and oscil-

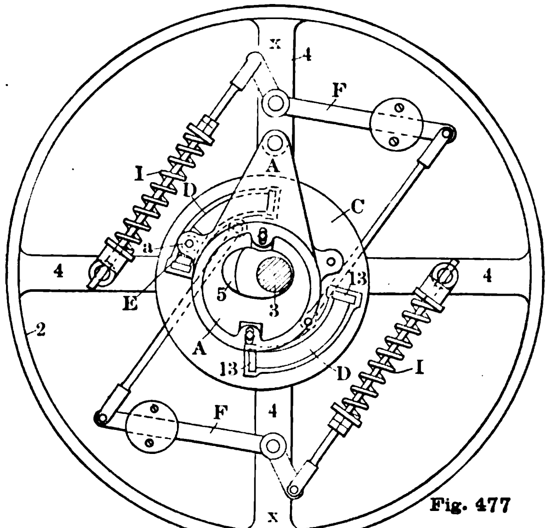


Fig. 477

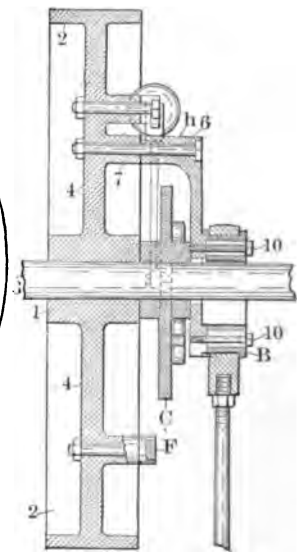


Fig. 479

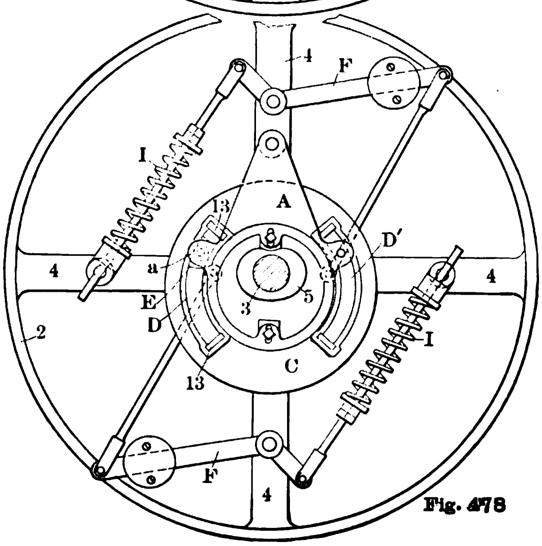


Fig. 478

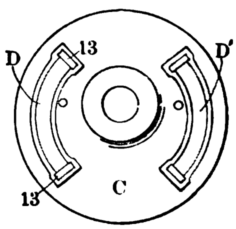


Fig. 480

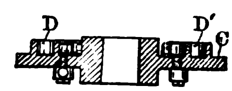


Fig. 481

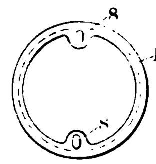


Fig. 482

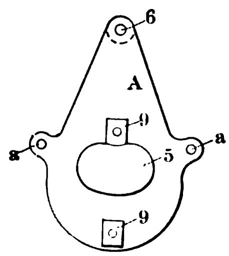


Fig. 483

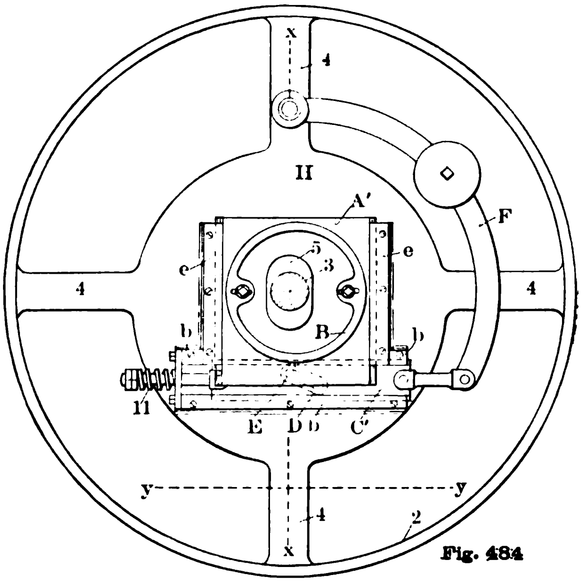


Fig. 484

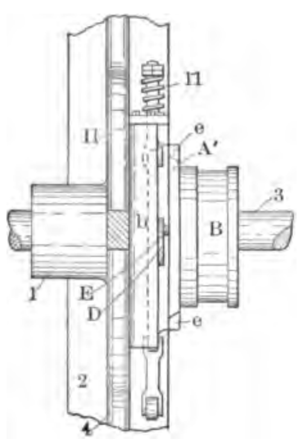


Fig. 486

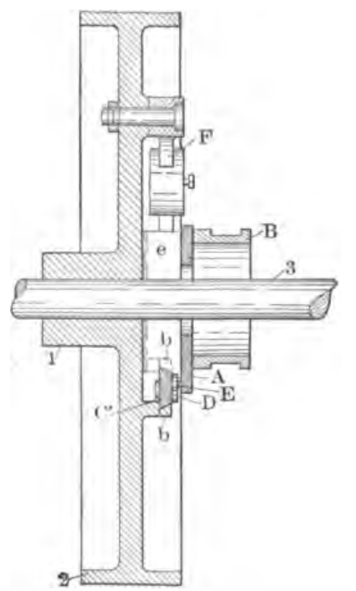


Fig. 485

late or move the plate C, which moves the slide E forward in the slot D, swinging the pendulum across the shaft and changing the position of the parts from that shown in Fig. 477 to that shown in Fig. 478. When the speed decreases the springs will cause the opposite movement of said parts.

Fig. 484 is a modification of Fig. 479, as previously stated, and shows a governor constructed with but one weighted arm F. The lever plate C′ is mounted in a guideway supported upon the enlarged hub or plate H; *b b* represent the guide in which the lever plate C′ moves. The eccentric supporting plate A′, instead of a pendulous lever, is supported by and moves in slides *e e*. The spring 11, which counterpoises the weighted arm in this instance, is attached to the opposite ends of the lever plate C′. These parts, A′ and C′, are the equivalents of the parts A and C shown in Figs. 477 and 478, except that they move in straight lines instead of curved. The weighted arm F, and spring 11, are the equivalents of the weighted arms and springs shown in Figs. 477 and 478, but are placed in different relation to act in substantially the same manner. It will be observed that the slide E moves a considerable distance in the guideway D to a comparatively short movement of the plate A′, owing to the slight eccentricity of the guideway; consequently the slide and guideway form a lock or a resistance sufficient to prevent the friction of the valve movement from affecting the operation of the governor.

It will be observed that the movement of the lever A (Figs. 477 and 478), or plate A′ (Fig. 484), is nearly at right lines to the path of the eccentric guideway D and the slide E traveling therein (Figs. 476, 479, and 484); consequently it makes it impossible for the movement of the valve to affect the governor, and yet the latter operates positively to move the plate A, or A′, and lengthen or shorten the valve travel. The plate C, when it is moved forward to arrest the increase of speed, travels in the same direction as the governor wheel and ahead of its speed. In order to prevent the bumping of the slide E, in case of extreme throw, elastic cushions 13 13 (Figs. 477 and 480) are provided for the ends of the guideways, against which the slide E will strike; but under ordinary circumstances the cushions are not required.

When the governor is to be changed, say from right to left, the slide E is detached from the guideway D, and placed in the guideway D′, and the levers and springs are reversed and occupy opposite quadrants of the wheel.

The main object of this construction is to provide a positive means for regulating the throw of the valve, and consequently the speed of the engine, which will readily move under slight variations of load and yet positively stop at the desired speed, avoiding oscillations or

tremor in changing under varying speed. This is accomplished by means of the traveling plate carrying a pin or slide engaging with the cam or eccentric guide for shifting the eccentric, which travels normally in right lines to the movement of the shifting plate, thereby locking the rapidly moving eccentric against the strain of the valve rod.

TO SET THE VALVES.

These valves, being slide valves, are set in a manner similar to that of ordinary slide valves, but as this engine contains marks peculiar to itself for setting the valves to avoid the necessity for removing the steam chest cover, an explanation will be necessary.

TO SET THE MAIN VALVE.

On the main valve stem will be found five marks, the center mark being 8 inches from the end of the steam chest; the two extreme marks represent the extreme travel of the valve, the two inner marks represent the lead or opening of the main valve port when the crank is on the dead center. This is not exactly the lead, as that is governed by the cut-off valve, and as this port is merely the passageway from the cut-off valve, it should have a large opening when the engine is on the center, so that the steam from the cut-off valve will have a free entrance.

For catching the marks on the valve stems, a trammel should be made of ¼ or $\frac{5}{16}$ inch square or round steel, with one end bent at right angle, about 2 or 2½ inches long, with end sharpened to a point, and the other end to be offset 1 to 1½ inches, parallel to the body of the trammel—the offset to be not over 2 inches in length, with end sharpened to a point. One end is placed in the center punch mark in the steam chest face, and the other is employed to catch the marks on the valve stems.

The crank is then placed on the dead center—if on the center toward the cylinder use the marks on the valve stem toward crank, and if on the outward center use the marks on stem nearest the cylinder. Then turn the eccentric in the opposite direction in which the engine is to run to a line inside of the 90 degree line, and tighten the set screw; and then turn the engine over on the other center and see if the opposite mark coincides with the trammel; if it does, the valve is correctly set; if it does not, equalize it by lengthening or shortening the eccentric rod until both are correct.

A line will be found on the shaft 90 degrees behind the crank, representing the normal line; and another will be found at an angular advance toward the crank. The center line of the eccentric has also a line marked on it, which, if placed to coincide with the angular

advance line, will be found to give the best general results, and also will coincide with the lines on the valve stem. Lines will also be found in just the reverse position for running in the opposite direction.

If the compression should have to be changed, move the eccentric forward or backward from this angular advance line, as may be required; this will change the marks, as they are made according to the compression for the average condition, power, and size of the engine, as engines are often subjected to conditions that require them to be changed accordingly.

TO SET CUT-OFF VALVES.

On the cut-off valve stem will be found five marks, the center mark being 8 inches from the face of stuffing box, similar to those of the main valve, the two outer marks represent the extreme travel of the valve like those on the main valve stem, the exceptions being that the two inner marks represent the valve line to line over the ports without any lead. This valve is set exactly like the main valve according to the marks; the only difference being that the inner marks represent the valve line to line instead of the opening of the valve. This is done for the reason that the opening or lead in the main valve is constant, while the opening or lead in the cut-off valve may require to be varied to suit the condition of the work. The lead is then obtained after the valve has been adjusted and equalized, by simply moving the eccentric forward or backward for more or less lead.

TO ADJUST THE GOVERNOR.

In placing the governor in position the pendulum is set squarely opposite, or 180 degrees to the crank. The pin on which the pendulum swings is to be in the opposite position to that of the crank pin, and to get the required lead the governor is moved around the shaft, the pendulum pin following behind the crank the same as the main eccentric shaft in the direction in which the engine is to run, lessening the angle of 180 degrees between the crank and the pendulum as may be required.

The lead can also be varied by sliding the eccentric forward or backward on the pendulum, giving a longer or shorter stroke to the cut-off valve. Any adjustment, such as lead, compression, cut off, etc., can be varied independently, and all adjustments of the valve can be made without taking off the steam chest cover, by simply using the marks on the valve stem.

The speed of the engine can be varied about 10 per cent. each way, by tightening the spring for increase of speed, or slackening the spring for decrease of speed. For any greater variations of speed it is preferable to change the weights and springs.

The governor can be set to run the engine in either direction. If it is desired to reverse the motion of the engine, it can be accomplished by taking off the weight arms and reversing them on their pins, and coupling up the connecting rods to the cam plate, and the spring rods in the opposite position, and then taking out the sliding shoe in the cam slot that connects the pendulum lug and placing it in the opposite slot, and setting the pin in the corresponding lug. The weight arms should be placed so that the weights will follow behind the pivot pin of the arm in whichever direction the engine is to run.

CHAPTER XXV.

SLIDE-VALVE ENGINES.

The main feature of all steam engines is the valve gear, and in this consists the chief difference in the various types of engines. It is therefore of the utmost importance that engineers should carefully study the anatomy of the various types of steam engines to enable them to acquire a thorough knowledge of the construction and operation of their valve gear; for without that knowledge no engineer can lay claim to proficiency in his profession, any more than a physician can lay claim to proficiency in *his* profession when he does not thoroughly understand the anatomy of the human body.

In order, then, to enable engineers to acquire the knowledge so essential to a successful prosecution of their profession, it has been the purpose and aim of the author to lay down a course of instruction upon the construction and operation of the valve gear of the various distinctly different types of steam engines, in the simplest form, and yet embody all the information necessary to enable engineers to acquire a thoroughly scientific, as well as practical, knowledge of the valve motion of steam engines generally. Therefore, to complete the course of instruction in regard to stationary engines, the slide-valve engine will receive consideration here.

The slide-valve engine is the pioneer of modern steam engines, and no other type of engine has ever gone into such universal use. Its construction and operation are therefore better understood among engineers, generally, than the construction and operation of any other type of steam engines. But as no work on steam engineering would be complete without embracing the slide-valve engine, this chapter will be devoted to an explanation and illustration of the main features of this particular engine, taking for such explanation and illustration two high-class and well-known makes, beginning with

THE SINKER-DAVIS SLIDE-VALVE ENGINE.

This engine is provided with a balanced valve, as shown in Fig. 487, which is a longitudinal elevation partly in section.

Fig. 488 is a plan view of the engine.

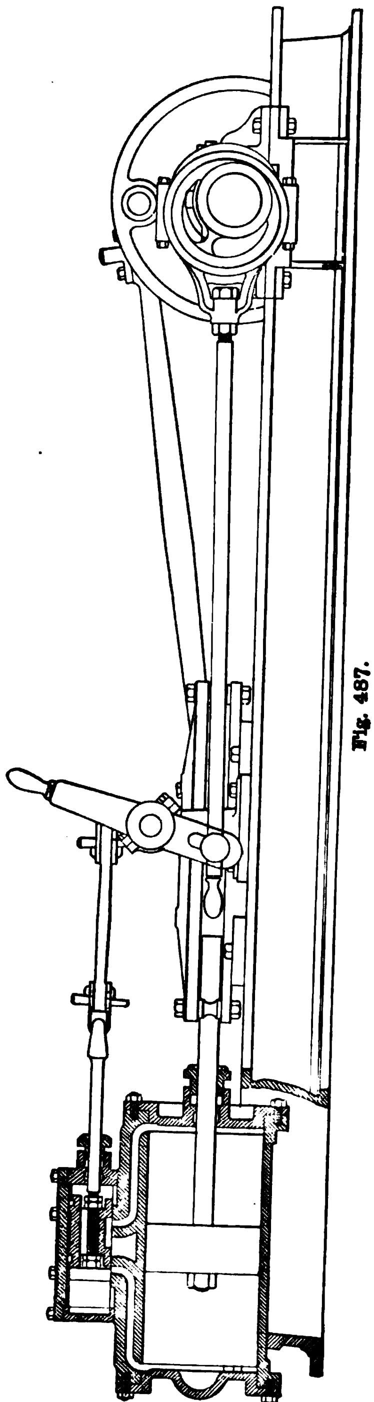


Fig. 487.

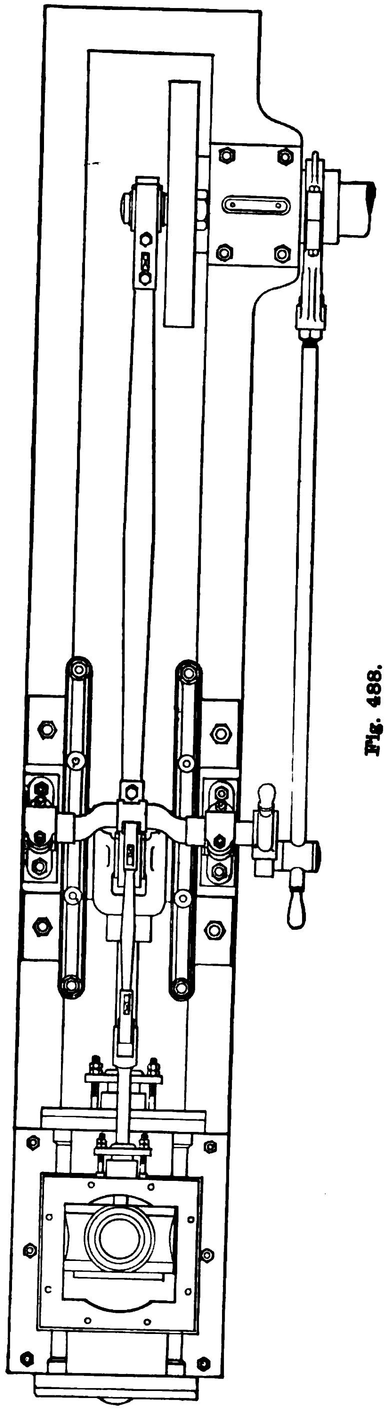


Fig. 488.

VALVE AND CRANK MOVEMENT.

Fig. 489 represents the position of the valve at the beginning of the forward stroke, and shows the lead opening.

Fig. 490 shows the position of the crank when the steam port P^1 is full open.

Fig. 491 shows the forward position of the crank when the steam port P^1 is closed, and also the point of the stroke at which expansion commences.

Fig. 492 shows the position of the crank at which the valve closes the exhaust, and the point of the stroke at which compression commences on the opposite side of the cylinder. In this position port P^2 is closed.

Fig. 493 shows the port P^1 open to exhaust, and the port P^2 open to lead.

Fig. 494 shows port P^1 full open to the backward or return movement of the crank pin.

Fig. 495 is a diagram showing the various positions of the crank pin corresponding to the various positions shown in Figs. 489, 490, 491, 492, 493 and 494.

TO SET THE VALVES.

First. Put the engine on the dead center.

Second. Place the valve central over the ports.

Third. Place the rock shaft in a perpendicular position and connect valve rod without moving valve or rock shaft.

Fourth. Place the throw of the eccentric down on the shaft, and adjust the nuts on the eccentric rod until the cam hook will drop onto the rock shaft pin.

Fifth. Turn the eccentric around on the main shaft, and adjust the nuts on the eccentric rod until the valve travels an equal distance on the outer side of each of the steam ports.

Sixth. Turn the eccentric in the direction toward the crank pin in which the engine is to run, until the valve has $\frac{1}{32}$ inch opening, and then tighten the set screws in the eccentric.

Seventh. Revolve the main shaft and put the engine on the other dead center, and note whether or not the lead is the same as it was on the original dead center, if so, the valve gear is correctly adjusted. If there is a difference, continue putting the engine from one dead center to the other and adjusting the eccentric rod until the lead is equal on both sides.

The eccentric in this engine follows the crank. When putting the engine on the dead center revolve the main shaft in the direction oppo-

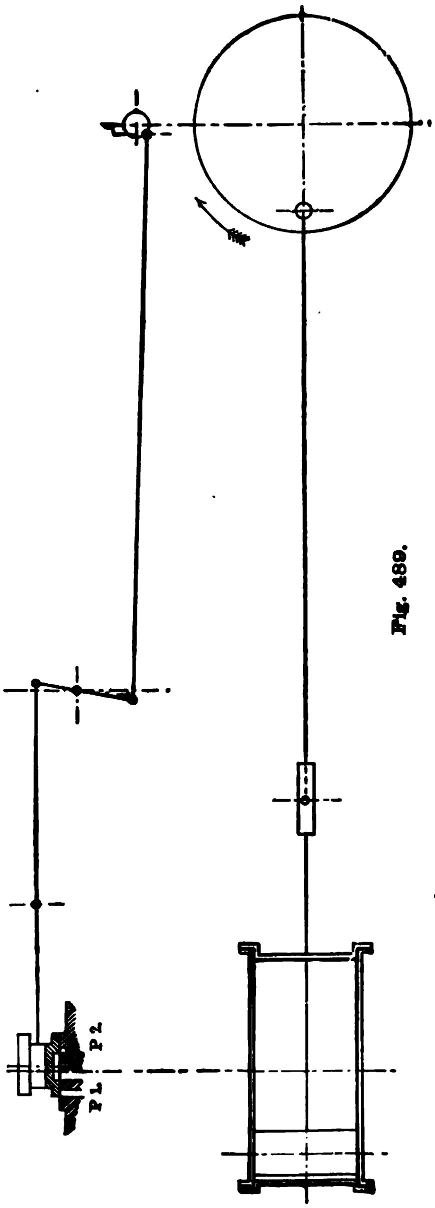


Fig. 489.

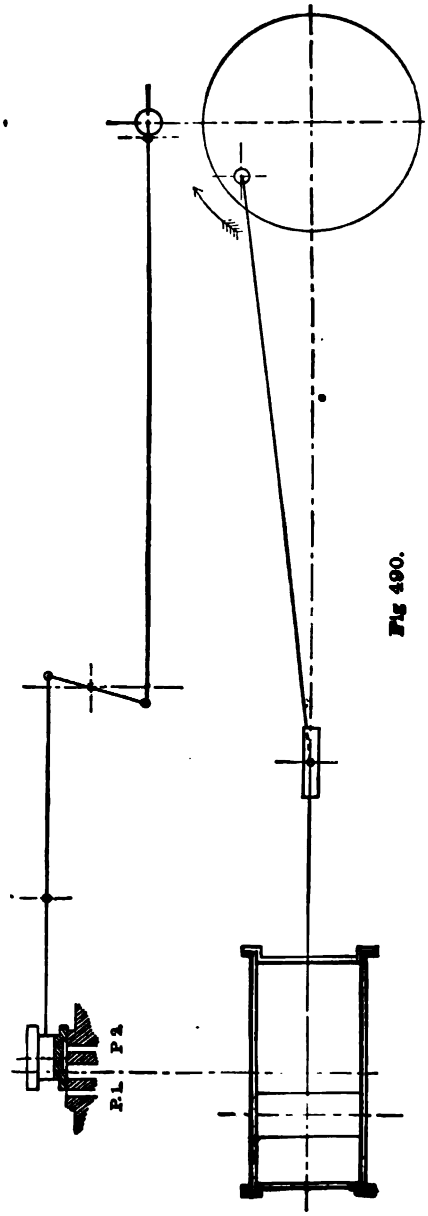


Fig 490.

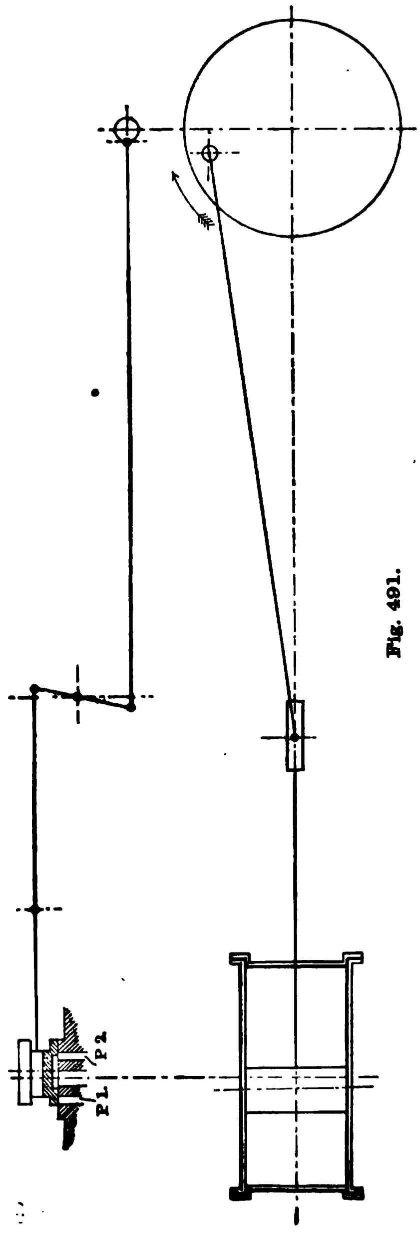


Fig. 491.

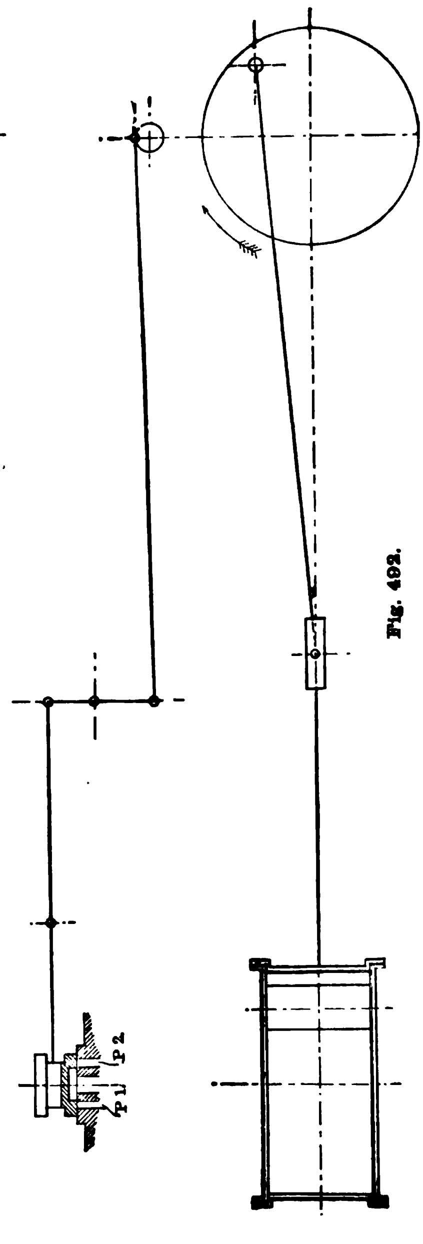


Fig. 492.

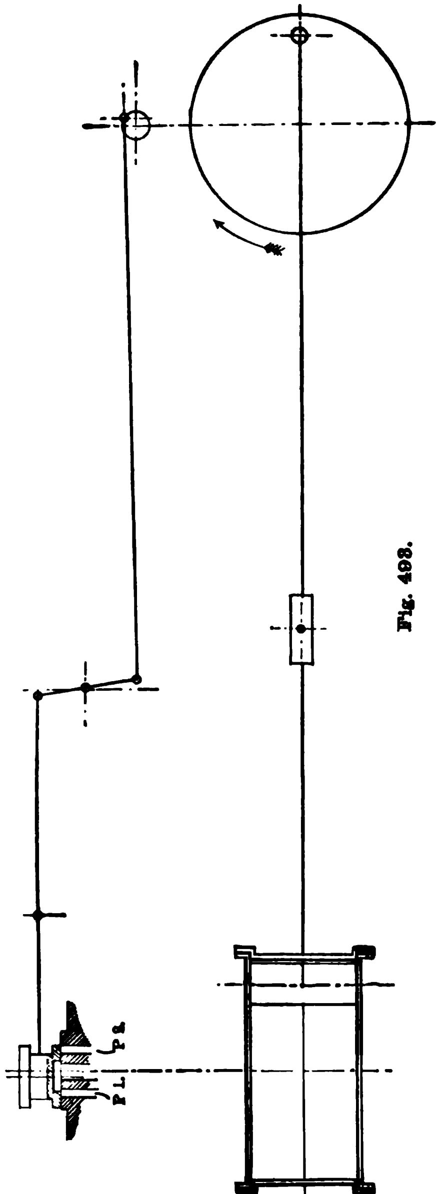


Fig. 493.

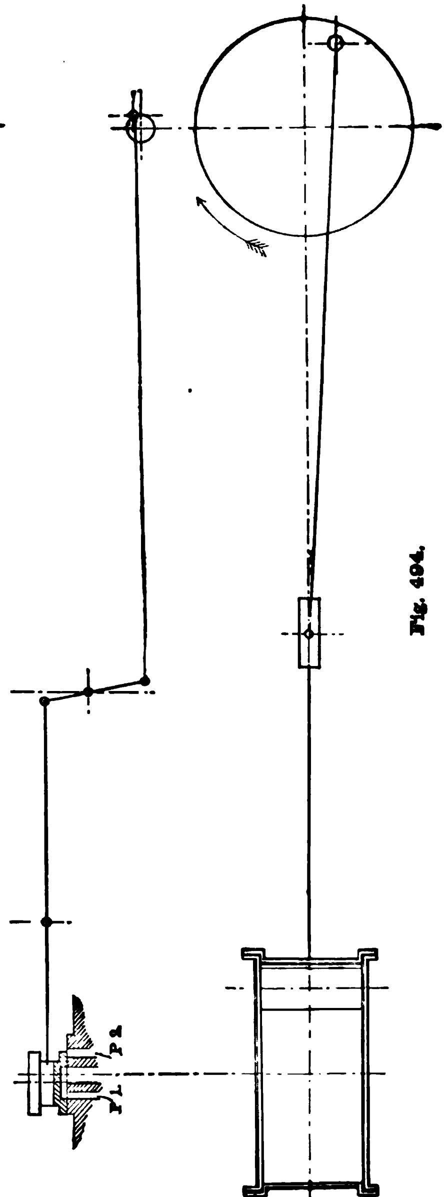


Fig. 494.

site to that in which the engine is to run, and never allow the crank pin to pass the dead center. If the crank pin should pass the dead center, turn the main shaft back about one-quarter turn and come ahead again until the crank pin comes fairly to the dead center, that will take up any lost motion that may exist in any of the working parts and bring the valve to its proper position.

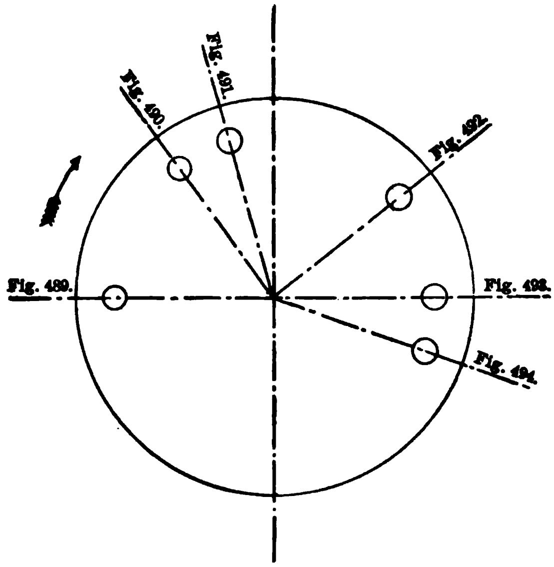


Fig. 495.

THROTTLING SLIDE-VALVE NON-CONDENSING COMPOUND ENGINES.

The slide valve has long been recognized as the simplest instrument for distributing steam to the cylinder of a steam engine. It has never been considered as economical as an automatic cut-off gear when applied to single cylinders, chiefly because it can not be made to satisfactorily cut off steam much earlier than one-half stroke, thus limiting the number of expansions of steam to about two, whereas the expansions found most economical in non-condensing simple engines range from three to five, according as stroke is short or long. Slide-valve engines have not until recently been constructed to carry high mean effective pressures, the poor grade of workmanship, design, and material common to many of them being such that anything over 25 pounds mean effective pressure would seriously strain the machine. On the other hand, with stronger parts and better material and workmanship, so that 40 pounds or 50 pounds mean effective pressure can be maintained, a better economy has been secured, although not as good as that of the best long stroke four-valve automatics.

ADVANTAGES OF THE SLIDE VALVE.

The slide valve of itself possesses two prominent advantages—it gives easy relief to water, especially if the valve is located at the side of the cylinder; and it can be made and kept tight very easily, thus preventing the serious loss of leakage common to many other styles of valve gear.

With the advent of the compound and triple expansion engines, the chief limitation to the usefulness of the slide valve has disappeared, for the number of expansions does not now depend altogether upon the cut off, but chiefly upon the ratio of the cylinder areas. For instance, if two cylinders are employed, one two and a half times the volume of the other, and steam be cut off on the smaller at one-half stroke, we obtain by this means about five expansions, and the result can be accomplished by means of slide valves, one for each cylinder. This principle lies at the base of the modern marine engine design where cut offs of five-eighths and one-half are used, and yet with triple and quadruple expansion engines fourteen and twenty expansions are obtained.

ADVANTAGES OF SLIDE-VALVE NON-CONDENSING COMPOUND ENGINES.

When applied to non-condensing compound engines the slide valve, in combination with a throttling governor, possesses—first, one important advantage over simple automatic engines; and second, another important advantage over the automatic compounds.

The first advantage consists in the fact that it can work under light loads—that is, with small mean effective pressures—more economically than any simple automatic engine, especially when its high pressure cylinder is steam jacketed. There are four reasons for this: First—A smaller mean effective pressure can be obtained by means of the expansion due to a fixed cut off of a throttling compound, without expanding below atmospheric pressure, than by means of automatic gears in which small mean effective pressures are secured by early cut offs, and the expansion extends below the atmosphere, accompanied by a great loss of power and an expenditure of steam. Second—Small mean effective pressures can be obtained by moderately late fixed cut off and a throttling governor without an increase in the relative amount of cylinder condensation. With automatic cut offs the small mean effective pressures developed by early cut offs are attended by increased percentages of cylinder condensation. Third—A steam jacket on the high pressure cylinder of a throttling compound is most effective in checking cylinder condensation with small mean effective pressures, as at that time there is the greatest difference in pressure and temperature between steam inside of cylinder and the steam at boiler pressure

in the jacket. Fourth—Clearance losses are less with the throttling compound than with automatic cut offs under light load; for with the former the ratio of clearance to volume of cylinder at cut off is constant at all loads, and the clearances are filled with steam at the throttle pressure only and from a back pressure equal to receiver pressure; with the latter the ratio of clearance to volume of cylinder at cut off becomes larger the earlier the cut off, and the clearances are filled with steam at boiler pressure and from a back pressure equal to the atmospheric pressure.

The second important advantage possessed by throttling compound over automatic compound non-condensing engines, lies in the fact that the power can be almost evenly divided between two cylinders, and the total maximum strain upon pistons is approximately proportional to the power developed. There are some automatic compounds which have the valve gears of both cylinders under control of the governor, so that the work of the cylinders is quite evenly divided, but the strains thrown upon the journals and reciprocating parts are as great, and often greater, under light loads than under heavy loads. The even division of load obtained by a throttle that regulates the initial and receiver pressures in proportion, and that produces small mean effective pressures without expansion below the atmosphere, permits the employment of the cross compound type of engine, which has been found impracticable with automatic cut off compounds, in which, with light loads, the low pressure cylinder is converted into a pump, and becomes noisy and difficult to operate.

THE HOUSTON, STANWOOD & GAMBLE CROSS COMPOUND ENGINES.

ADVANTAGES OF CROSS COMPOUND ENGINES.

The advantages possessed by a cross compound engine with cranks at right angles, where each engine performs its share of the work, are important They chiefly consist in great steadiness of motion, as the fly wheel does not have to carry load when crank pins pass their centers; small fluctuation of speed under sudden changes of load, as nearly the entire inertia of fly wheel is available to resist rapid external disturbances; strains are divided between two engines and are not concentrated in a single machine; speed can be reduced as much as 50 per cent. on account of high efficiency of fly wheel, so that large engines may be installed at low speeds for temporary light loads; in case of repair of either high or low pressure engine, the one or the other may be operated singly under control of governor at half load; and finally pistons are accessible and are easily removed, which is not the case with the tandem engines.

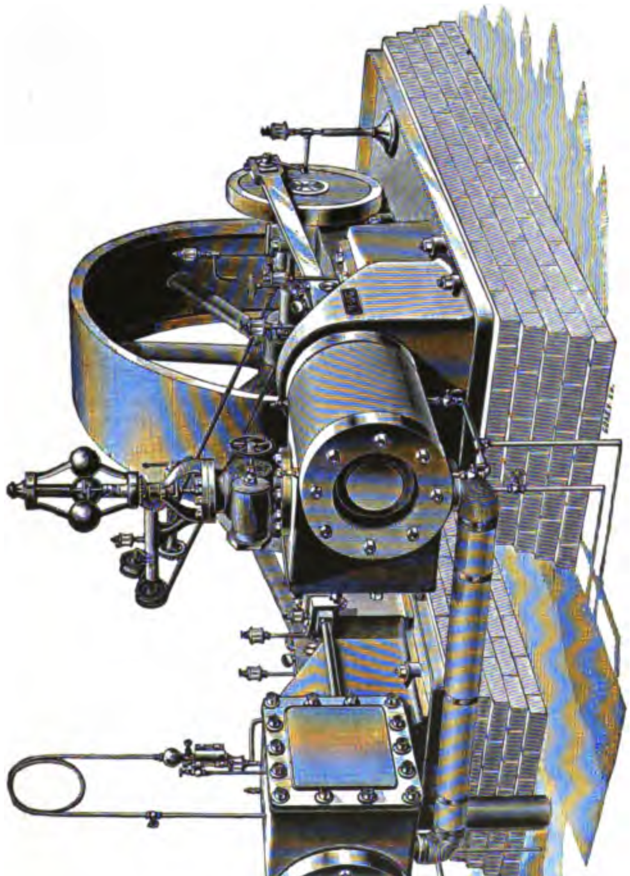


Fig. 496.

HOUSTON, STANWOOD & GAMBLE CROSS COMPOUND ENGINE.

Figs. 497 and 498 are diagrams taken from the high and low pressure cylinders of a 200 horse power engine of this type; they show how evenly the power is divided between both cylinders. These diagrams when combined into a single block diagram, as shown in the chapter on Indicator Practice, will show the action of steam, supposing the expansions to have all taken place in the low pressure cylinder only. This arrangement of diagrams indicates the losses that occur by the passage of steam from one cylinder to the other. It also shows the small amount of power required to fill both high and low pressure clearance spaces with steam by compression, as compared with a single cylinder.

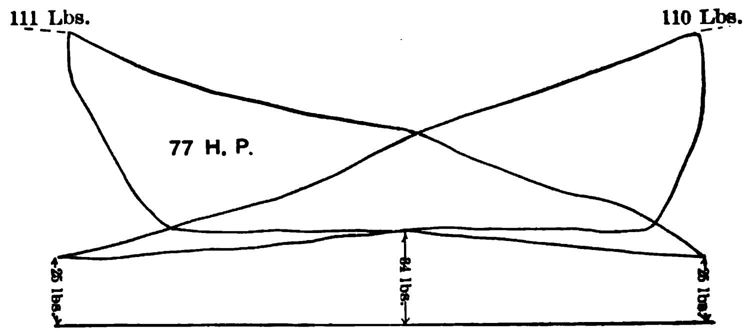


Fig. 497.

HIGH PRESSURE CYLINDER—60 SPRING,

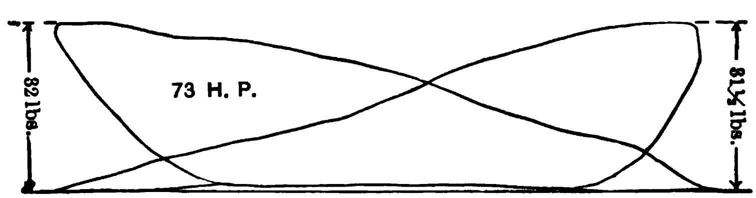


Fig. 498.

LOW PRESSURE CYLINDER—30 SPRING.

TO FIND THE HORSE POWER.

To find the horse power of these engines, indicator diagrams have to be taken from each cylinder; the horse power of each cylinder is calculated as if each cylinder were a simple engine, and the sum of horse powers is the total power of the engine. Thus, multiply the mean effective pressure found by diagram of the high pressure cylinder, by its area in square inches, by its piston speed in feet per minute, and divide the product thus found by 33,000, the quotient equals horse power developed by high pressure engine; then multiply mean effective pressure found by diagram of the low pressure cylinder, by its area in

square inches, by its piston speed in feet per minute, and divide the product thus found by 33,000, the quotient equals horse power developed by low pressure engine. In Figs. 497 and 498 these results are 77 for high pressure engine and 73 for low pressure engine, or total horse power equals $77+73=150$ indicated horse power, or the load under which engine was working when diagrams were taken.

To determine the maximum horse power that engines of this type develop, the builders have arranged a series of numbers which have been chosen for each of the thirty-three sizes they build, so that if the number representing the size of a given engine be multiplied by the boiler pressure, and by the number of revolutions and the product divided by 10,000, the quotient will give the maximum indicated horse power of the engine.

Example.—What horse power will engine No. 49 develop with 80 pounds of steam at 170 revolutions?

$$49\times80=3920;\ 3920\times170=666400$$

$$\frac{666400}{10000}=66.64 \text{ Horse power.}$$

TO SET THE VALVES.

The valves are set in each engine as if each was a plain slide-valve engine. Houston, Stanwood & Gamble have marks on their engines which will assist the engineer in this work. The shaft and eccentrics are marked so that eccentrics can be located correctly for engines to run either "over or under." The valves have from $\frac{1}{16}$ inch to $\frac{1}{8}$ inch lead, according to the size of the engine. To ascertain if valve has proper opening, there are tram marks on the crank disc by which the crank can be placed exactly on center. To do this, select a piece of thick telegraph wire, about 18 inches long, sharpen ends to a point, and bend 2 inches of same square at both ends to make a trammel 14 inches from point to point. Place the engine with any crank nearly on center by the eye, then search on corner lug of outside foundation bolt for a trammel point; from it set the other point of trammel in the point easily found on edge of crank disc, at a distance of 14 inches by the wire tram. When this is done the engine is exactly on its center. This can be done for all the centers. When cranks are thus placed the lead can be examined and equalized, so that the opening is the same for all dead center positions of cranks.

TO SET THE VALVES OF SLIDE-VALVE ENGINES GENERALLY.

First. The steam chest cover being removed, and the valve stem and eccentric rod being disconnected, place the valve V exactly in the

center of its travel, as shown in Fig. 499; that is, place it so that the edge of the valve at one end will extend over the outer edge of the steam port S at that end, exactly the same distance that the edge of the valve at the other end does over the outer edge of that steam port.

Second. Place the rocker arm R straight up and down, and adjust the valve stem V S so that it will connect with the valve and rocker arm without moving either from the positions in which they have been placed, and then connect the valve stem to the valve and rocker arm without moving either of them.

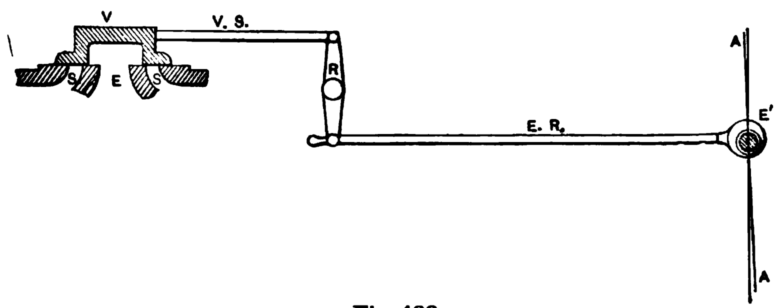


Fig. 499.

Third. Turn the eccentric E′ loosely around the main shaft until its throw will be at a right angle with the eccentric rod when connected to the rocker arm, as shown by line A A, and then adjust the eccentric rod so that it will connect with the rocker arm without moving the rocker arm or the eccentric.

Fourth. With the eccentric rod and valve stem connected, as shown in Fig. 499, turn the eccentric around the main shaft and note the distance the valve at each end has uncovered the steam ports, and if the travel of the valve from the outer edge of each steam port toward the exhaust port is equal, the eccentric rod is properly adjusted. If the travel of the valve, as described, is not equal at both ends, lengthen or shorten the eccentric rod, as may be required, and again turn the eccentric around the shaft, and if there still appears to be a difference in the travel of the valve at each end, lengthen or shorten the eccentric rod, as may be required, and continue the operation of revolving the eccentric and adjusting the eccentric rod until the valve uncovers each steam port exactly the same width.

Fifth. If the travel of the valve is too great lower the eccentric-rod pin of the rocker arm; if the travel of the valve is too short raise the eccentric-rod pin of the rocker arm, and continue these adjustments until the travel of the valve is correct.

Sixth. Place the engine on either dead center. But for simple and convenient description we will begin by placing the engine on the dead center toward the cylinder, and if the engine is to run under on the outward stroke, turn the eccentric around the shaft in the direction in which the engine is to run, until the eccentric stands up at right angle with the crank; then move the eccentric toward the cylinder and downward toward the crank pin until the valve at the piston end of the cylinder has the desired lead, and then fasten the eccentric securely in position.

Seventh. If the engine is to run over on the outward stroke, and the engine is on the dead center toward the cylinder, turn the eccentric around the main shaft, in the direction in which the engine is to run, until the eccentric hangs down at right angle with the crank; then turn the eccentric toward the cylinder and upward toward the crank pin until the valve at the piston end of the cylinder has the desired lead, and then fasten the eccentric securely in position.

Eighth. Place the engine on the other dead center, and if the lead of the valve is the same as it was on the first dead center, the adjustment is properly made. If the lead is not the same move the eccentric one-half the difference in the required direction, and adjust the eccentric rod the other half of the difference, and continue putting the engine from one dead center to the other, and making adjustments by moving the eccentric and adjusting the eccentric rod until the lead of the valve is equal at both ends.

If the engine is put exactly on the dead center in the first instance, there will be no difficulty in setting the valve properly if the eccentric rod was properly adjusted, and the valve and rocker arm properly placed to begin with. Care must also be taken during subsequent operations to get the engine exactly on the dead center, as any failure to get the engine on the dead center will make it impossible to set the valve correctly.

CHAPTER XXVI.

LOCOMOTIVES.

The science of steam engineering applies to locomotives as well as it does to stationary or marine engines. The same knowledge of that science requisite to make a stationary or a marine engineer proficient in his calling is just as indispensable to the completion of the education of a locomotive engineer as it is to those who follow the other two branches of steam engineering. It is, therefore, unnecessary to repeat in this chapter anything which has preceded it. There are some things, however, which belong peculiarly to the profession of locomotive engineering, not met with in either of the other branches of steam engineering science. Therefore, to complete the course of instruction laid down in this work, and to make it a library of steam engineering in fact as well as in name, that part of the science applicable chiefly to locomotive engineering will be treated in this chapter.

The chief feature of this branch of steam engineering, and the one in which locomotive engineers are, as a general rule, mostly interested, is that of valve setting. In order, then, that the instructions here given may be easily and readily understood, they will be given in their simplest form possible. But to secure accuracy in performing the work, the student is informed that the greatest care must be exercised in putting the engine on the dead center, for the reason that unless that is accurately done, all subsequent movements in the setting of the valves, or in the adjustment of any other part of the valve gear, will be incorrectly performed. There are various methods employed for putting locomotives on the dead center, but few of which insure the accuracy so indispensable to correct valve setting. Therefore, the instructions here contained will embody the most accurate methods known to the profession.

HOW TO SET THE VALVES OF A LOCOMOTIVE.

PUTTING THE ENGINE ON THE DEAD CENTER.

The right or left-hand engine may be the first to be put on the dead center, and they may be put on either the forward or backward centers; but for the sake of simplicity and convenient description the right-hand engine will be put on the forward dead center to begin

with. But before proceeding to do that disconnect both parallel rods, unless there is a perfectly level track on which to perform the operation; and then set up the forward driving box wedges just tight enough to prevent any play of the driving boxes between the jaws of the frame. Then take up all the lost motion in the back and forward ends of the main rods, and the engine will be ready to put on the dead center. The crank pins of locomotives are set at right angles to each other, that is, the crank pins on one side are set at right angles to the crank pins on the other side. In some locomotives the left-hand crank pins lead the right-hand crank pins, but in this case we will assume that the right-hand crank pins lead the left-hand crank pins. We will now proceed to show how to put the locomotive on the dead center.

First. To put the right-hand engine on its dead center, revolve the forward driving wheels forward until the cross head has traveled to within a short distance of the forward end of the stroke on the right-hand side; then put a straight mark on the cross head and continue it onto one of the guides.

Second. Before moving the driving wheels again put a similar mark on the cross head and guide of the left-hand engine.

Third. Return to the right-hand side of the locomotive and resume turning the driving wheels forward until the crank pin has passed over the dead center and moved the cross head forward to the end of its travel and back until the mark on the cross head and the mark on the guide of the right-hand engine come exactly line and line.

Fourth. Before moving the driving wheels again, and while the marks on the cross head and guide of the right-hand engine are line and line, proceed to the left-hand side of the locomotive and put a mark on the guide of the left-hand engine, so that it will be in line with the mark already on the cross head. Then find the center between the two marks on the guide on that side and place a center mark there similar to the two already on the guide.

Fifth. Remain on the left-hand side of the locomotive and turn the driving wheels backward until the mark on the cross head of the left-hand engine comes squarely up to the center mark on the guide on that side, and the right-hand engine will then be practically on the dead center—the difference in the angularity of the rod when the cross head is at either end marks and the center mark, alone prevents absolute accuracy. The angularity of the rod when the first mark is made on the cross head and guide is exactly the same as it is when the second mark was made; but as the wrist pin of the driving wheel moves in a circle, there is a very slight difference in the angularity of the rod when the mark on the cross head is placed midway between the two

outside marks; but the length of main rods is generally such that the difference between the results obtained by this method and absolute accuracy is hardly perceptible; while the result generally obtained by the old method with marks on the rim of the driving wheel tire and wheel cover is uncertain; because the locomotive, being on springs, the wheel cover is liable to move up or down with the boiler every time the driving wheels are turned, and hence the mark on the wheel cover rarely, if ever, settles to the identical position occupied before the wheels were revolved.

PUTTING THE ENGINE ON THE DEAD CENTER—ANOTHER METHOD.

Another method, which if performed according to instructions here given, will insure perfect accuracy in putting the engine on the dead center, is as follows:

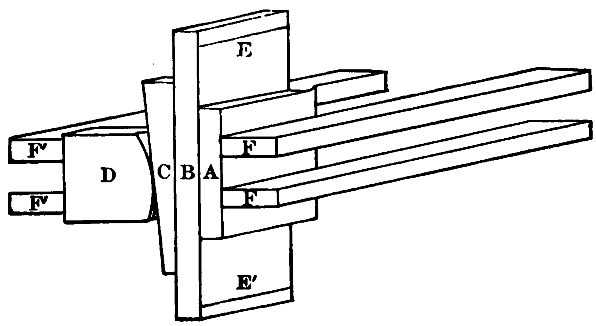


Fig. 500.

First. Disconnect both ends of the main and parallel rods, and move the cross head of each engine to the forward striking point, so that they will be out of the way for the operations described further on.

Second. Set up the wedges of the forward driving boxes just tight enough to prevent any play between the boxes and the jaws of the frames.

Third. Procure a planed board for each engine, of any convenient thickness, 3 or 4 inches in width, about 2 inches longer than half the length of the stroke of the engine, and having at least one straight edge to work from. Draw a line square across each end of the board, at such distance apart that the distance from the line E to the line E′, on the board B, as shown in Fig. 500, will be exactly equal to one-half the length of the stroke of the engine.

Fourth. Fasten the board B, with its face against the back of the board A, with the face of the board A against the inner edges of the outside guides, as shown in Fig. 500, or in any other convenient man-

ner; but so that the face of the board B will be in line with the middle of the forward crank pin. Which position, it will be observed, is determined by the thickness of the board A, or by a number of boards.

To determine the thickness of material to be placed between the face of the board B and the inner edges of the outer guides, in the space occupied by the board A, measure from the middle of the forward crank pin to the middle of the main rod journal; then measure the same distance from the middle of the cross head wrist pin toward the inner edge of the outer guides, and the distance between the end of that measurement and the inner edge of the outer guides will give the thickness of material required to be placed between the face of the board B and the inner edges of the outer guides. For example: If the length of the forward crank pin from the face of the hub of the driv-

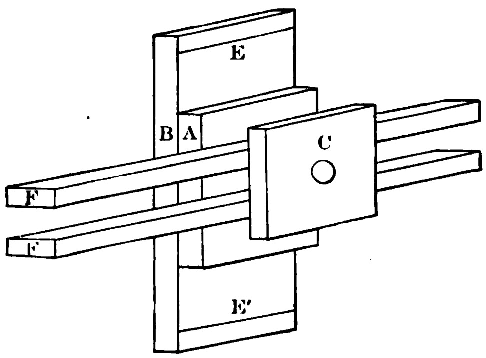


Fig. 501

ing wheel to the end of the pin is 10 inches, the middle of the pin will be 5 inches. If the distance from the middle of the main rod journal to the middle of the pin is, say 2 inches, then the distance from the middle of the cross head wrist pin toward the inner edge of the outer guides will be 2 inches, and the distance between those 2 inches and the inner edge of the upper outside guide will give the thickness of material required to be placed between the face of the board B and the inner edges of the outer guides; so that if the distance from the middle of the cross head wrist pin should be, say 4 inches, the thickness of material required to fill the space occupied by the board A (Fig. 500) will be just 2 inches.

After determining the required thickness of the board A, place it against the inner edges ef the outer guides F F, and as near the cross head as possible, and wedge it firmly in place by the aid of the wedge C and block D, against the inner edge of the inner guides F′ F′, as shown in Fig. 500, or by clamping the boards against the inner edges

of the outer guides F F, by a bolt passing through the boards A B C, as shown by the position of the boards in Fig. 501, or in any other convenient manner.

After fastening this board B, as shown in Figs. 500 and 501, with the face of the board in line with the middle of the forward crank pin, adjust the board so that it will be square with the guides, and that the lines E E′ will be equidistant from the center of the piston rod. Or, in other words, so that the line E will be the same distance above the top of the piston rod as the line E′ is below the bottom of the piston rod, and the board will be in its proper position. Should the board B be a quarter of an inch or so out of line with the middle of the forward crank pin, or if the lines E E′ be not exactly equidistant from the center of the piston rod, it will be near enough correct for practical purposes. But exactness should be the aim of all engineers.

Fifth. If a trammel can not be procured of the length here required, procure a board thick enough and wide enough to prevent much sagging at its center, and long enough to reach about one inch beyond the center of the board B, on the lines E E′, and one inch beyond the center of the forward driving axle. Then, if the centers in the forward driving axle and crank pins are large, plug them with lead, and with the aid of compasses, locate a small center mark at the center of each end of the axle, and at the center of each forward crank pin. Then take the long board, previously procured, and construct a temporary trammel by driving a round wire nail through each end, and at such distance apart as to reach from the center of the forward driving axle to somewhere near the center of the board B, on the lines E E′ (Figs. 500 and 501). Then put the point of the nail of one end of the trammel into the center of the forward driving axle, and press the point of the nail at the other end of the trammel into the board B at line E at the top, and then press the same nail into the board B at line E′ at the bottom with the point of the nail in the back end of the board in the center of the driving axle. Then perform the same operation on the other side of the engine.

We are now ready to put either side on the forward or back dead center, but for the sake of simplicity and convenient description, we will begin by putting the right-hand crank pin on the forward dead center; and to accomplish that proceed on the left-hand side of the locomotive, as follows:

First. Place the point of one of the nails in one end of the trammel into the small hole previously made in the board B at line E, and bring the point of the nail in the other end of the trammel to the center of the axle, to see if there has been any change. If there has been no change, and the point of the nail at the axle comes fairly into the center of the axle, raise that end of the trammel to where the center

of the crank pin in revolving will meet the point of the nail in the trammel, keeping the nail at the other end in the hole in the board B at line E, and cause the forward driving wheels to be revolved so that the crank pin will move upward until its center comes fairly in line with the point of the nail in the trammel at that end, and the crank pin on the right-hand side of the forward driving wheel will be exactly on the dead center.

Second. To put the left-hand forward crank pin on the dead center take the trammel to the right-hand side of the locomotive, and put the point of the nail at one end into the hole or center mark in the board B at line E′ at the lower end of the board below the guides, and bring the point of the nail at the other end into the center of the forward driving axle, to see if there has been any change. If there has been no change, and the point of the nail in the trammel at the axle end comes fairly into the center of the axle, lower that end of the trammel down to where the center of the crank pin in revolving will meet the point of the nail in the trammel, keeping the nail at the other end in the hole or center mark in the board B at line E′ at the lower end of the board, and cause the driving wheel to be turned forward until the crank pin center comes fairly to the point of the nail in the trammel at that end, and the crank pin of the forward driving wheel on the left-hand side of the locomotive will be exactly on the forward dead center.

Third. Return to the left-hand side with the trammel and place the point of one of the nails in the trammel into the hole or center previously made in the board B, at the line E′, below the guides, and bring the point of the nail at the other end of the trammel to the center of the forward driving axle to see if any change has taken place. If there has been any change, simply put the point of the nail at the axle to the center of the axle, and make another center in the board B, at the line E′, below the guides, with the nail at that end of the trammel; then keep the nail in that position, lower the other end of the trammel down to where the center of the crank pin in revolving will come in line with the point of the nail, and cause the forward driving wheels to be revolved forward until the center of the crank pin comes fairly to the point of the nail in the trammel at that end, and the crank pin of the forward driving wheel on the other side of the locomotive will be exactly and squarely on the back dead center.

Fourth Return to the right-hand side of the locomotive with the trammel and place the point of one of the nails in the trammel in the hole or center mark of the board B at the line E previously made above the guides, and bring the point of the nail at the other end of the trammel to the center of the forward driving axle, to see if any change has taken place. If there has been no change, and the point

of the nail at the axle end of the trammel comes fairly into the center of the axle, raise that end of the trammel so that the center of the crank pin in revolving will meet the point of the nail in the trammel. keeping the nail at the other end in the hole or center in the board B. at the line E, above the guides, and cause the driving wheels to be turned forward until the center of the crank pin comes fairly to the point of the nail in the trammel at that end, and the crank pin of the forward driving wheel on the left-hand side of the locomotive will be exactly and squarely on the back dead center.

This concludes the operation of putting the engine on dead centers; but in the foregoing description it has been assumed that the right-hand crank pin leads the left-hand crank pin. Still, it does not make a particle of difference in the operation of placing crank pins on the dead center. In cases where the left-hand crank pin leads the right-hand crank pin, when the right-hand crank pin is up and at right angle with the line of travel of the piston, the left-hand crank pin is on the forward dead center. In cases where the right-hand crank pin leads the left-hand crank pin, and the right-hand crank pin is up and at right angle with the line of travel of the piston, the left-hand crank pin is on the back dead center. It is, therefore, obvious that where one crank pin leads the other, as is always the case in locomotives, the other is always one-quarter of a revolution behind the one that leads, so that whenever the crank pin that leads is on the forward dead center the other crank pin is up and at right angle with the line of the piston travel; if the crank pin that leads is on the back dead center, the crank pin on the other side is down and at right angle with the line of the piston travel. If the crank pin that follows is on the forward dead center, the crank pin on the other side is down and at right angle with the line of the piston travel; if the crank pin that follows is on the back dead center the crank pin on the other side is up and at right angle with the travel of the piston.

We are now prepared to take the first step in locomotive valve setting.

TO ADJUST LOCOMOTIVE VALVE GEAR.

While operations may be begun on either side of the locomotive, yet, for the sake of simplicity and convenient description, we will begin each of the series of operations on the right-hand side, after disconnecting all of the driving rods, eccentric blades and valve stems and removing steam chest covers, and setting up the wedges for forward driving boxes so as to take up all lost motion in driving boxes; taking care to get them just tight enough to prevent any forward or backward movement of the driving boxes between the jaws of the frames.

This being done, we are prepared to begin adjustment of the valve gear.

First. Place the center of the right-hand valve exactly in the middle of its travel, that is with the center between the two steam ports, having first placed the reverse lever in the center notch of the quadrant.

Second. Observe whether or not the rocker arm is exactly at right angle with the guides; if it is not, make adjustments so that it will be exactly at right angle with the guides, no matter whether the guides are horizontal or whether they incline little or much downward toward the center of the driving axle.

Third. After having placed the rocker arm at right angle with the guides, connect the valve stem with the rocker arm without moving the valve from its central position, or without moving the rocker arm from its central position.

Fourth. Place the link in its neutral position at right angles with a line drawn through the center of the axle and center of the lower rocker-arm pin, and fasten the link in that position.

Fifth. Turn the forward motion eccentric loosely around the axle, in the direction the engine is to run, until the eccentric has been turned upward and stands at right angle with the eccentric blade when connected to the link; fasten the eccentric in that position temporarily; then adjust the eccentric blade and connect it with the link without moving the link or eccentric.

Sixth. Turn the backward motion eccentric loosely around the axle, in the direction the engine is to run, until the eccentric has been turned downward and stands at right angle with the eccentric blade when connected with the link; fasten the eccentric in that position temporarily; then adjust the eccentric blade and connect it with the link without moving the link or eccentric.

Seventh. Place the reverse lever in the forward notch of the quadrant, and loosen the forward eccentric and turn it completely around the axle, and note the forward and backward extreme travel of the valve. If the valve has traveled an equal distance on both sides of the outer edge of each outer steam port, the adjustment is correct. If the valve has traveled too far forward, lengthen the eccentric blade, if the valve has traveled too far backward shorten the eccentric blade. For example, if the valve has traveled half an inch too far forward, lengthen the eccentric blade half an inch; if the valve has traveled half an inch too far backward, shorten the eccentric blade half an inch. Keep up these adjustments until the eccentric, in being revolved loosely around the axle, will cause the valve to travel an equal distance on each side of the ports.

When these adjustments have been completed for the forward motion eccentric, place the reverse lever in the back notch of the quadrant and repeat, with the backward motion eccentric, the operation described for the forward motion eccentric, until the travel of the valve on both sides of the ports has been equalized for the motion of the backward eccentric. Then, to equalize the valve travel on the left-hand side of the locomotive, repeat the above described operation for both left-hand eccentrics.

Eighth. Put the crank pin of the forward right-hand driving wheel on the forward dead center; place the reverse lever in the forward notch of the quadrant. Place the forward eccentric straight up and the backward eccentric straight down, then turn the upper part of the forward eccentric forward and downward toward the crank pin until the valve has the desired lead for going head, and fasten the eccentric in that position.

Ninth. Place the reverse lever in the back notch of the quadrant, and turn the lower part of the backward eccentric forward and upward toward the crank pin on the forward center until the valve has the desired lead for backing, and fasten the eccentric in position.

Tenth. Put the crank pin of the forward left-hand driving wheel on the forward dead center; place the reverse lever in the forward notch of the quadrant; place the forward eccentric straight up and the backward eccentric straight down; then turn the upper part of the forward eccentric forward and downward toward the crank pin until the valve has the desired lead for engine going ahead, then fasten the eccentric in that position.

Eleventh. Place the reverse lever in the back notch of the quadrant, and turn the lower part of the backward eccentric forward and upward toward the crank pin on the forward center until the valve has the desired lead for engine backing, then fasten the eccentric in that position.

Twelfth. Put the crank pin of the forward right-hand driving wheel on the back dead center, and place the reverse lever in the forward notch of the quadrant, and notice if the lead of the valve is the same as it was when the crank pin was on the forward dead center; if it is, the adjustment is right; if it is not, divide the difference by lengthening or shortening, as may be required, the blade of the forward eccentric one-half of the difference and move the eccentric the other half of the difference, and fasten the eccentric in that position.

Thirteenth. Place the reverse lever in the back notch of the quadrant, and notice if the lead of the valve is the same as it was when the crank pin was on the forward dead center; if it is, the adjustment is right; if it is not, divide the difference by lengthening or

shortening, as may be required, the blade of the backward eccentric one-half the difference, and then moving the backing eccentric the other half of the difference.

Fourteenth. Put the crank pin of the forward left-hand driving wheel on the back dead center, and place the reverse lever in the forward notch of the quadrant, and notice if the lead of the valve is the same as it was on the forward dead center; if it is, the adjustment is right; if it is not, adjust the eccentric blade and move the eccentric according to directions given in twelve and thirteen.

Fifteenth. Place the reverse lever in the back notch of the quadrant, and notice if the lead of the valves is the same as it was on the forward dead center; if it is, the adjustment is right; if it is not, adjust the eccentric blade and move the eccentric according to directions given in twelve and thirteen.

If the valve gear was readjusted on either side of the locomotive, when the crank pin was on the back dead center, that side should again be put on the forward dead center, to see if the lead is right; if it is not, the adjustments must continue until it has been made right; that is, the engine must be put from one dead center to the other until the valve lead has been equalized. If the valve lead when engines were on back centers was the same as when they were on the forward centers, it will not be necessary to again put them on the forward dead centers; but all parts of the valve gear should be properly secured to place, so that there will be no likelihood of any of the parts getting loose when the locomotive is under way. This completes the valve setting.

The steam chest cover can now be replaced and the main rods connected. But before the parallel rods are connected the back wedges should be set up, and the axles and crank pins properly trammed; after which the parallel rods can be connected, and the locomotive will be ready for service.

TO ADJUST ECCENTRICS BEFORE DRIVING WHEELS ARE PLACED UNDER A LOCOMOTIVE.

Adjusting the eccentrics when the driving wheels are under the locomotive, and the valve gear is properly connected, and one engine is on the dead center, has already been thoroughly explained; it will therefore be the purpose here to explain how to adjust locomotive eccentrics to their proper positions before the driving wheels are put under the locomotives.

The first thing to be done is to find the diameter of the inside collars of the crank pin of the driving wheel attached to the axle on which

the eccentrics are to be placed; then scribe a circle on the end of the axle, making the diameter of the circle the same as the diameter of the inside collar of the crank pin. Then revolve the driving wheels until the inside collar of the crank pin and the circle on the axle come in line horizontally, using a spirit level to get them in line accurately. The spirit level is placed on top of the collar of the crank pin, and on top of the circle on the end of the axle, and the wheels are adjusted until the spirit level in that position comes to a perfect level, as shown in Fig. 502, when the wheels are blocked in place for the adjustment of the eccentrics on that end of the axle.

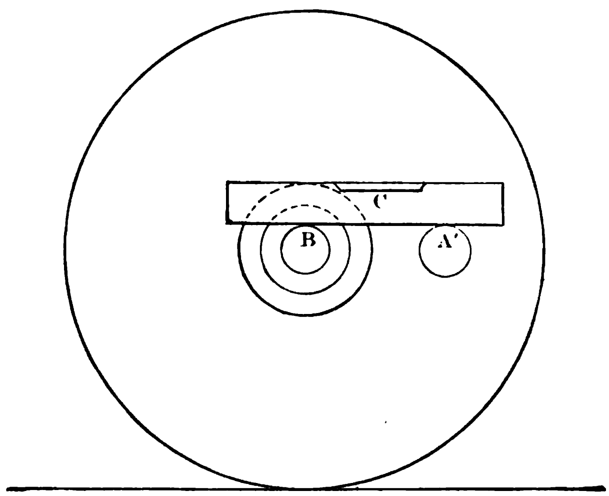


Fig. 502.

A′ represents the inside collar of the crank pin; B the circle on the end of of the axle, corresponding to the diameter of the crank pin collar; and C a spirit level resting on top of crank pin collar in line with top of circle B.

In adjusting the eccentrics according to this method, the driving wheels should have their crank pins put on the forward center, as shown in Fig. 502, and the eccentrics on that end of the axle should be adjusted with the driving wheels in that position, after which the crank on the other or left-hand side should be put on the forward center.

The next thing to be done is to procure a board E, as shown in Figs. 503 and 504, with parallel edges, thick enough to set on edge without side support; and one of such edges to be true. Before putting the board in position, as shown in the illustration, lay off the diameter of the driving axle and the diameter of the eccentrics on the

board near its middle. Make a center mark in the board at A, and from that lay off one-half of the diameter of the axle on each side of that center, and one-half of the diameter of the eccentric on each side of the center, so that the two inner marks at the ends of the arrows A will represent the diameter of the axle, and the marks at the ends of the arrows B will represent the diameter of the eccentric. If there is any difference in the diameters of the eccentrics, the diameter of each should appear on the board; but eccentrics should always be made equal in diameter and in throw.

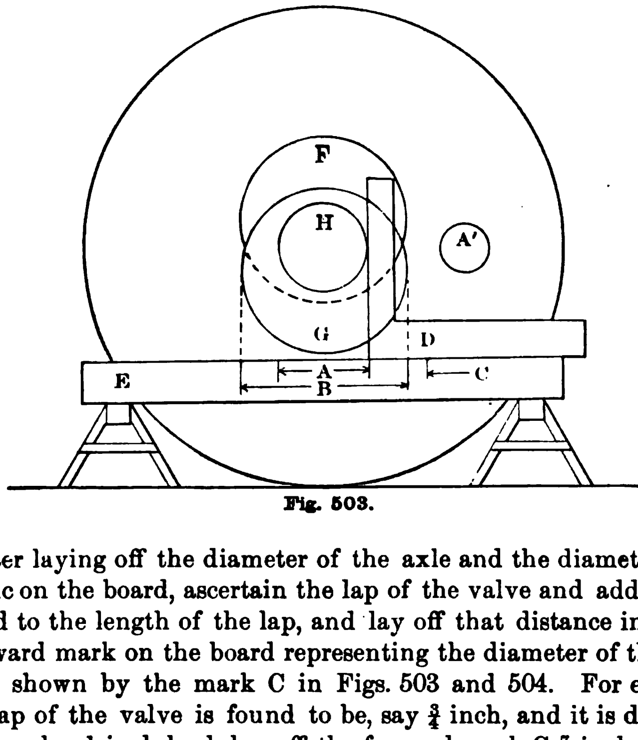


Fig. 503.

After laying off the diameter of the axle and the diameter of the eccentric on the board, ascertain the lap of the valve and add the lead required to the length of the lap, and lay off that distance in front of the forward mark on the board representing the diameter of the eccentrics, as shown by the mark C in Figs. 503 and 504. For example: If the lap of the valve is found to be, say $\frac{3}{4}$ inch, and it is desired to give the valve $\frac{1}{8}$ inch lead, lay off the forward mark C $\frac{7}{8}$ inch from the forward mark B, and that distance will represent the lap and lead of the valve.

Next, place the board under the eccentric, with the true edge of board up, in position as shown in the illustration, low enough to clear the eccentric when down. Level the board with a spirit level, move the eccentric to one side out of the way for the time being, and place a square on top of the board E, with the corner of the square at the forward mark A, and move the board toward the axle until the upper part of the square touches the axle, then the marks on the board, representing the diameter of the axle, will be squared with the axle.

If the board is under the forward eccentric turn the eccentric up; if it is under the backing eccentric, turn that eccentric down, and keep it there temporarily. H represents the axle, F the go-ahead eccentric, G the backing eccentric, A the crank pin, and D the square.

Now place the square D, as shown in Fig. 504, in line with the lap and lead mark C. If the board is under the backing eccentric, move the eccentric forward and upward until it touches the square, then put a mark on the axle and eccentric with the eccentric in that position, to mark the proper position of the eccentric; then fasten the eccentric securely in that position, taking care to see that the eccentric is the proper distance from the inside of the hub of the driving wheel.

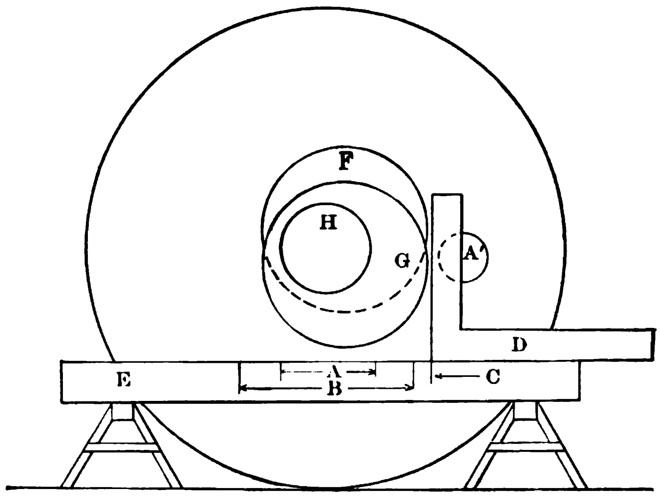


Fig. 504.

Next, move the board over to and under the forward or "go ahead" eccentric, and move the eccentric out of the way for the time being. Level the board and get the marks on the board, representing the diameter of the axle squarely in line with the sides of the axle. Bring the eccentric to its proper place on the axle and turn it into an upright position. Place the square at the lap and lead mark on the board and move the eccentric forward and downward toward the crank pin until the eccentric touches the square; fasten the eccentric in that position, and put a mark on the axle and eccentric to mark the proper position of the eccentric.

Repeat the above operation for the opposite side and the eccentrics will be properly adjusted.

TO ADJUST LOCOMOTIVE VALVE GEAR IN CASE OF DERANGEMENT ON THE ROAD.

In case the valve gear of a locomotive becomes deranged while the engine is out on the road, it is of the utmost importance that the engineer should understand how to determine the cause, how to locate it, and how to apply the remedy quickly. It will, therefore, be the purpose of these instructions to impart the information necessary in such cases.

As soon as any derangement occurs in the valve gear when the locomotive is under way, it will be indicated by irregular exhaust, and the engine should be brought to a stop at once; and when brought to a stop the reverse lever should be thrown into the back notch of the quadrant, and the throttle valve opened and the engine backed. If the engine stops after backing, or refuses to back, it is a pretty sure sign that something about the valve gear is broken. If the engine backs and stops on the center on either side, or refuses to start to backing from the center on either side, the cause is pretty sure to exist on the opposite side, and an examination may show a broken arm or a broken valve stem. In which event the valve on that side should be placed squarely over the ports, and if the rocker arm is broken, the eccentric straps, rods, and link should be removed, and the main rod should be disconnected. But in no case should the parallel rod be disconnected unless the opposite parallel rod, or one of its straps, is broken; in which event, both parallel rods should be disconnected. If the valve stem is broken, disconnect it from the rocker arm; place the valve squarely over the ports; disconnect the main rod on that side; pinch the engine over the center, and go ahead.

If the engine backs all right, start the engine ahead, and if it stops on the center on one side, it will indicate that the trouble is on the other side, and that the forward eccentric has slipped. Pinch the engine, on the side on which the eccentric has slipped, onto the forward center, and throw the reverse lever into the back notch in the quadrant, and put a mark on the valve stem just outside of the gland of the stuffing box. This mark will indicate that the valve has uncovered the forward steam port just to the extent of the lead of the valve. Now throw the reverse lever into the forward notch of the quadrant and the mark made on the valve stem will disappear in the stuffing box gland, or move outward from its face. Now move the forward eccentric loosely around the axle in the direction the engine is set to run, until the mark on the valve stem appears at the face of the stuffing box gland, and fasten the eccentric firmly in that position; the valve will now be properly set, and the engine will be in condition to go ahead.

COMPOUND LOCOMOTIVES.

Figs. 505, 506, 507, 508 and 509, represent the Weir-Harden compound locomotive. The only difference between this and ordinary locomotives is the cylinder, valve, and steam chest; all of which can be attached to any ordinary locomotive without changing or altering any other part of the valve gear. The cylinder is a continuous casting, as shown in Fig. 509. This cylinder is cast with an annular partition ring or bridge wall at the center, of sufficient thickness to withstand the steam pressure, and to allow a packing ring to be inserted therein. The partition ring is made of such depth as to give the differential area between the high and low pressure piston surfaces, the latter having an area of about three to that of one for the high pressure surface. The piston is made in three parts—two followers and a cylindrical shell or piston barrel interposed between them. It will thus be seen that there are four piston faces in one cylinder—two of a small annular area for high pressure steam, and two of a larger area for low pressure steam. There are three packing rings in each follower and one in the center partition ring or bridge wall. These rings are of the Harris type of two thin rings riveted together, and then returned and cut, with a lap joint, into six or eight sections, each of which is provided with a special spring set into a pocket just large enough to hold and guide it at each joint. A small brass casting is set into each spring to give it a bearing at the joint of the ring. In the partition ring or bridge wall the springs are placed in the reverse position from that of the follower packing rings, so as to close the rings around the piston barrel. The piston rod passes through both followers and the piston barrel, and is provided with a shoulder and steel collar at the rear and two nuts at the forward follower, which constitute the means for tightening up the piston parts and locking them securely in place. It will thus be seen that this presents greater simplicity of arrangement than that presented in the ordinary piston. Beside, the followers being some distance apart, together with the partition ring or bridge wall, form three bearings, and thus providing a most excellent guide for the piston, giving it a smooth running effect, and preventing the plunging or gouging on the cylinder surfaces, especially in the case of large pistons, thus obviating the necessity of passing the piston rod through the forward cylinder head to avoid the difficulties encountered in many large engines.

The piston barrel is subjected only to external pressure, and may be made very light on that account. The followers can also be made very light on account of their peculiar construction, so that the whole is but a trifle heavier than the ordinary piston. Another important feature in this style of engines is the absence of unequal strain to

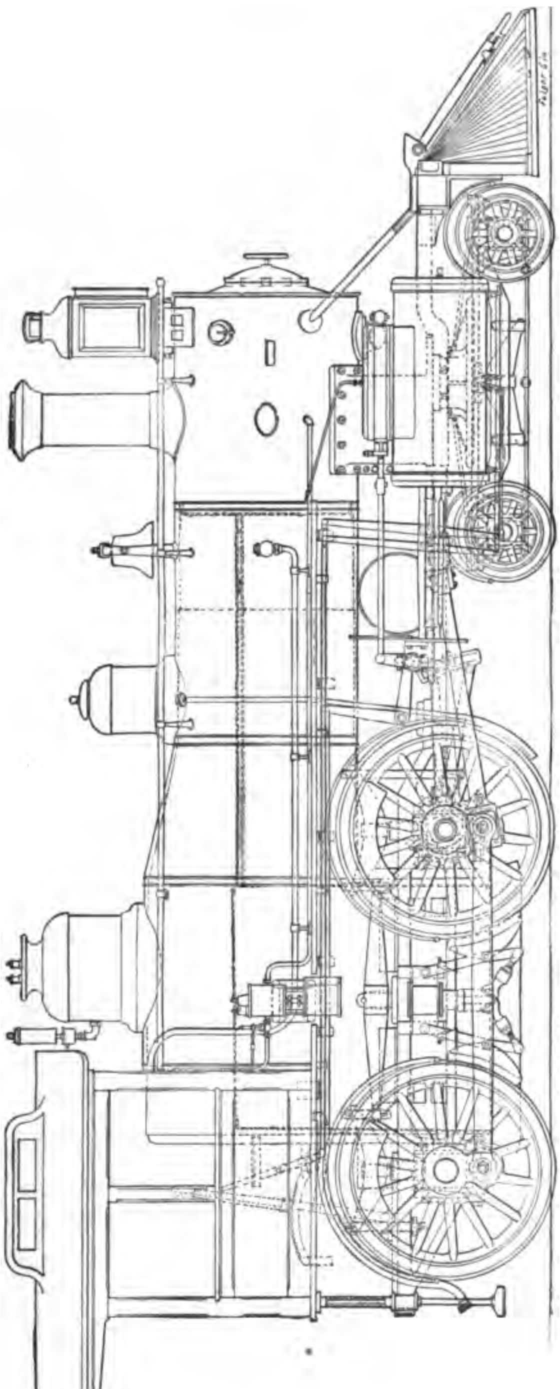


Fig. 505.

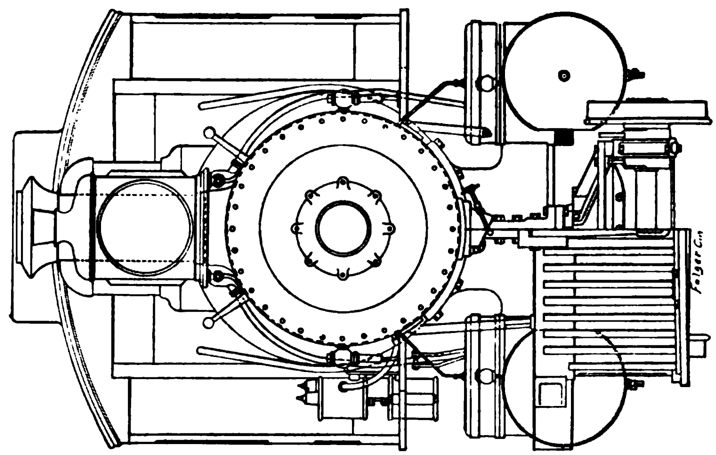


Fig. 508.
FRONT ELEVATION. | FRONT ELEVATION, PILOT REMOVED.
LOOKING BACKWARD.

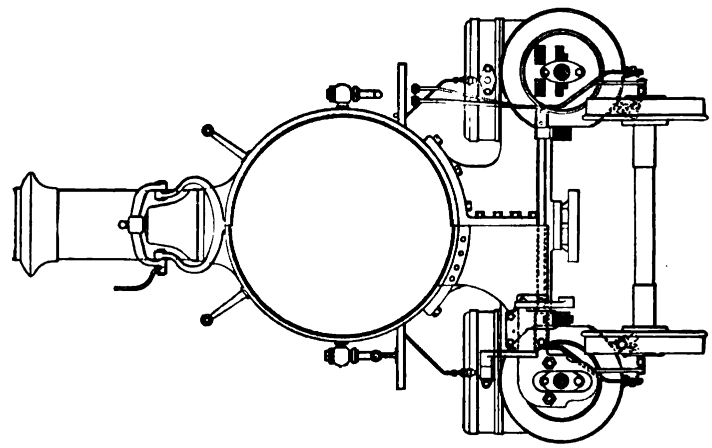


Fig. 507.
SECTION BACK OF ROCK SHAFT. | SECTION FRONT OF ROCK SHAFT.
LOOKING FORWARD.

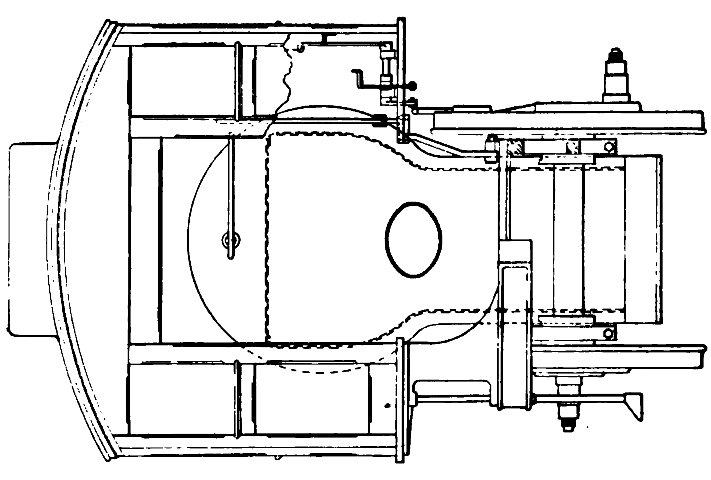


Fig. 506.
REAR ELEVATION. | SECTION SHOWING POSITION OF LEVERS.
LOOKING FORWARD.

which the pistons of ordinary compound engines are subjected, especially where they are connected by separate connecting rods to their respective cranks.

This style of engine has also the advantage of having but one stuffing box, instead of from three to four as in some types of compound engines. Again, the oil for lubricating the interior of the cylinder is admitted with the high pressure steam, and lubricates the high and low pressure cylinders at the same time, as the high and low pressure cylinders are one and the same in this engine.

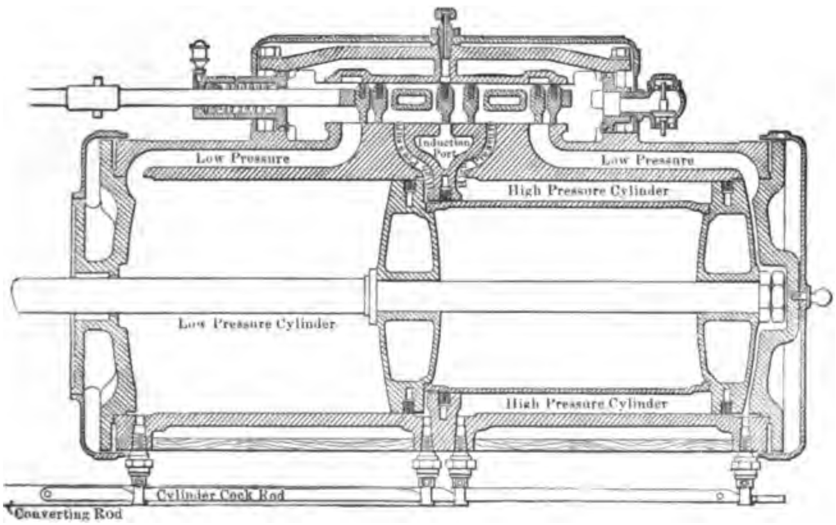


Fig. 509.

Another feature of this engine is that the low pressure steam performs its work in the same space that the high pressure steam performs its work; hence it is not subjected to the evils of condensation that the steam in the ordinary type of low pressure engines is subjected to, because the walls of the cylinder are kept at a higher temperature. Another feature of this engine is the shortness of the distance the steam has to travel from the high to the low pressure surface of the piston. The ports are very large in area, and are proportioned according to the area of the respective piston surfaces and the maximum speed at which the engine is to run, yet, they being exceedingly short, the clearance is kept at a minimum.

The induction port is located in the sweep of the cylinder where the exhaust port is commonly located in simple engines. The exhaust ports are also located in the sweep of the cylinder, one at each end of the seat, or where the induction port is located in ordinary slide-valve

engines. The exhaust steam leads to a large exhaust opening in the saddle which gives it a free exit.

The steam chest contains only the exhaust steam, as the live steam is admitted through the induction port under the valve. The steam chest forms a passage for the exhaust steam to the exhaust ports in the saddle. The steam chest cover has a plate cast to its under side, which forms the balance plate or upper seat of the slide valve. This plate has recesses cast in it opposite the ports for counterbalancing. These recesses have ribs cast across them flush with the seat to keep the packing strips in the valve from springing in.

The valve is of the slide-valve type, made in skeleton form, having the same area on top as on the bottom, except that the area of the upper portion of the valve to which the steam has access is slightly in excess of the lower portion subjected to steam pressure; the difference being provided for the purpose of keeping the valve to its seat; and yet the valve is so constructed as to present a very small surface either at its top or bottom. Each port controlling section is double or pierced by an auxiliary port, so that when any port is uncovered or open $\frac{1}{16}$ or $\frac{1}{8}$ inch the top of the valve opens that same amount, thereby doubling the amount of opening as compared with the actual travel of the valve. This action is performed on the steam induction port for the admission to the high pressure piston, and also for the release from the high and admission to the low pressure ports, as well as the exhaust from the low pressure ports, thereby giving large openings. This same action also balances the valve, by letting steam into the recesses in the steam chest cover. When the valve is cut back to its finest cut off all the ports are wide open on center except the steam inlet, which has less than a full port opening, but double that of the ordinary valve operated by link motion. The high pressure steam is admitted through the induction port under the valve alternately to each recess connecting with the high pressure port, thence transferring it to the low pressure port, and from there exhausting over the under side of the outer edge of the valve into the chest.

The upper face of the valve contains packing strips, the outer edge of each is opposite to and corresponds with the port edge on the lower seat, and they are held up in position by light springs; in addition thereto live steam is admitted under them by a series of holes leading from the center auxiliary port.

The valve is arranged loosely between the upper and lower seat so as to lift if necessary; but that feature is of no consequence in reversing the engine, as the valve is so constructed in connection with the ports that the steam can pass from one port to another or out into the chest, thus rendering it impossible to cause any damage.

CONVERTING VALVES.

The converting mechanism consists of a short U pipe, as shown in the cross section of cylinder (Fig. 510) each end of which connects with one of the high pressure ends of the cylinder at the center partition. Each of these pipes is provided with a quick opening gate valve, the whole occupying but a very small space, under and back of the cylinder, so as to be protected against being hit when the locomotive is under way; at the same time they are so situated as to be easily accessible. The mechanism for operating these valves consists of a small lever situated in the cab in front of the seat box, convenient to the

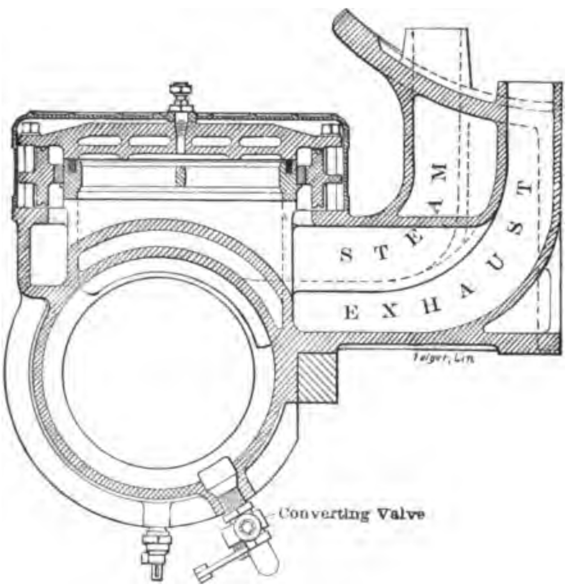


Fig. 510.

engineer, and connected by rods and arms to the converting valves. By the use of this lever the engine can be converted from a simple to a compound engine, and *vice versa*, at the pleasure of the engineer.

The method of converting the engine from a compound to a simple engine is as follows:

When the converting valve is opened the high pressure steam is admitted from the side of the piston which is using it through the converting valve and into the other high pressure part of the cylinder; and, as the main valve is in communication with the high and low pressure ports of that end of the cylinder, as it would be if working compound and using low pressure steam, the high pressure steam is acting on the large or low pressure part of the piston at that end; and that part of the piston having an area three times as great as that of

the high pressure part of the piston, the engine is given greatly increased power. The opening through the converting valve is of sufficient area to maintain boiler pressure in the low pressure part of the cylinder at slow speed.

VALVE GEAR.

The shifting link motion, as shown in Fig. 511, or any other valve gear employed on simple or ordinary locomotives, can be used without change except that the eccentrics are required to be reversed, with the position of crank pin on forward center, as shown at A (Fig. 511), or, in other words, set as if it were a direct-acting engine without the intervention of any rocker arm. With this single exception the valves and valve gear are adjusted and set in the same manner precisely as those of an ordinary locomotive. Hence the instructions relating to valve setting given in the forepart of this chapter are as applicable to this locomotive as they are to ordinary locomotives.

THE MAIN VALVE.

Fig. 512 is a longitudinal section through the center of the chest, valve and upper and lower seats. A A represents the valve seat of the cylinder, showing location of ports; B B, steam chest; C, steam chest cover, with plate D cast thereon, forming the upper or balance seat; E E, main valve; F, valve yoke around the valve; G, vacuum valve to relieve the cylinder while rolling without steam; H, valve stem packing case; I I I I I I, packing strips in the upper face of the valve. There is also one strip running lengthwise on each side of the valve, not shown in the illustration, into which the ends of the cross strips are set, and forming a steam tight joint on top of the valve. J, oil cup for valve stem; K, oil plug through which the oil is supplied to the valve from the lubricator in the cab; L, valve stem metallic packing; M, valve stem; N, induction port in the saddle of the cylinder, through which the cylinder is supplied with live steam through the valve, as shown by the arrows; O O, steam ports leading to the high pressure parts of the piston; P P, steam ports leading to the low pressure parts of the piston; Q Q, passages in slide valve through which the steam passes from the induction port N to the high pressure ports O O, and from ports O O to low pressure ports P P; R, auxiliary port for live steam, to give the double supply over the top of the valve through recess T, for passages Q Q; S S, auxiliary ports for giving the double supply of high and low pressure steam over the top of the valve through U U. The auxiliary ports R S S and recesses T U U are also for balancing the valve, by constantly keeping the steam over the valve on the same area as that which is exposed to the action of steam underneath. The recesses T U U have ribs level with the surface of the seat, to prevent the packing strips from jumping

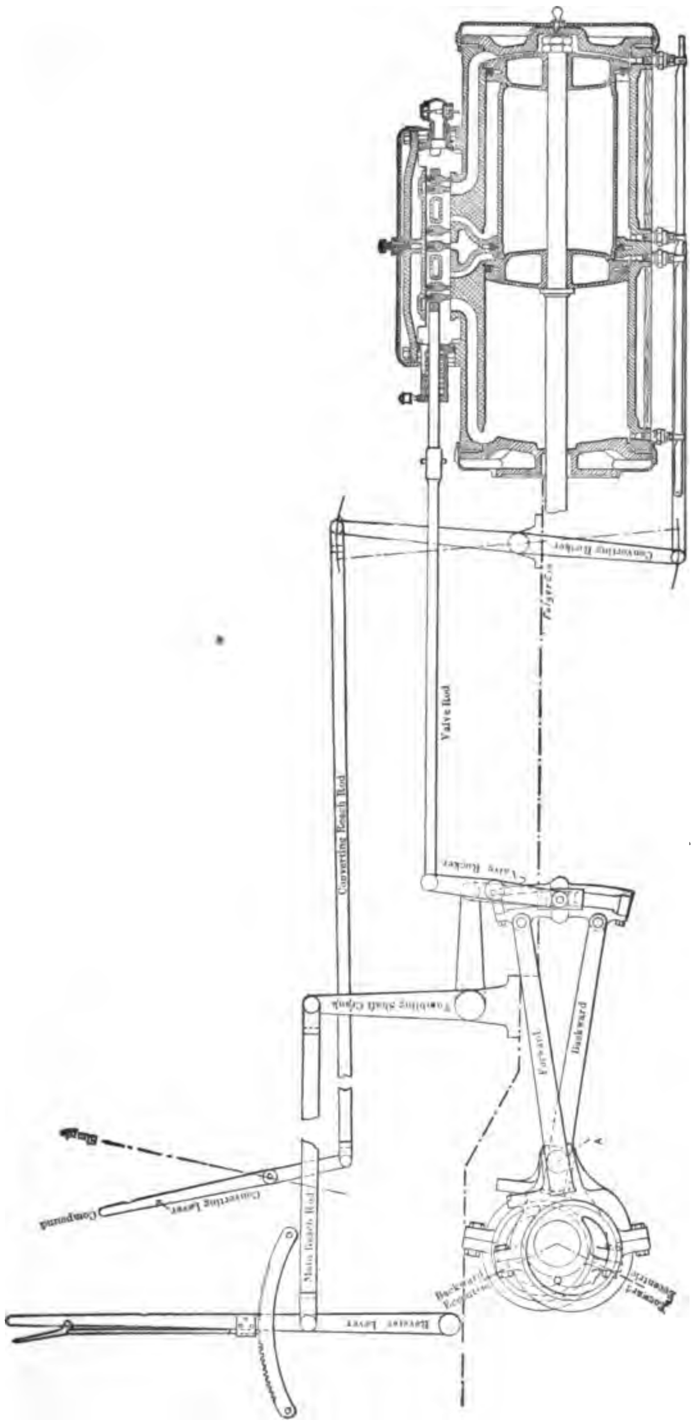

Converting Rocker
Valve Rod
Converting Reach Rod
Valve Rocker
Tumbling Shaft Crank
Forward
Backward
Converting Lever
Compound
Main Reach Rod
Reverse Lever
Backward Eccentric
Forward Eccentric

Fig. 511.

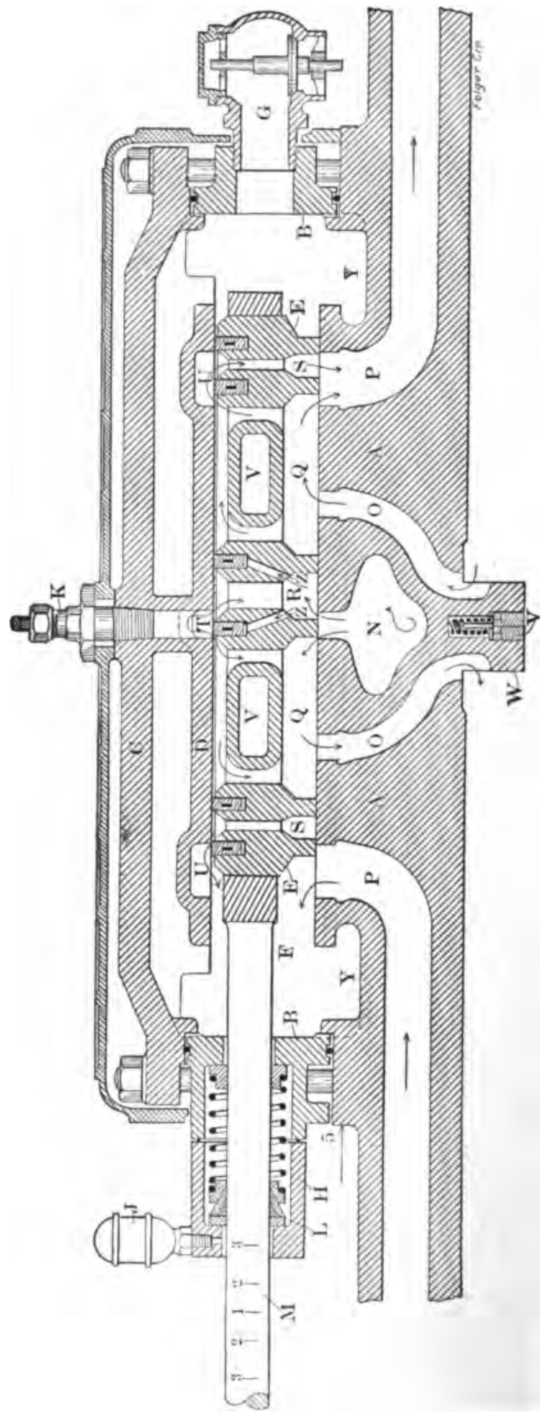


Fig. 512.

into them. V V, blocks cast in the valve, with a lightning core through each; these blocks are made large for the purpose of taking up space to reduce the amount of clearance; W, bridge wall or partition ring in the high pressure part of the cylinder; X, bridge wall packing which surrounds the barrel of the piston; Y Y, cavity formed in the cylinder and sweeping around the same, forming a passage for the exhaust steam to the saddle and into the exhaust pipe; Z Z, holes drilled from the live steam auxiliary port R to the under side of the packing strips, to admit live steam to the strips to hold them tight against the upper seat.

MARKS ON VALVE STEM FOR VALVE SETTING.

The marks 1 2 2 3 3 on the valve stem represent the usual marks put on by which to set the valve; which is done by an L-shaped trammel, the long end of which is placed against the point shown by the arrow at 5. The trammel is made to reach from 5 to 1 when the valve is in its central position over the ports. 2 2 represent the lap and lead of the valve for each direction from 1. 3 3 represent the extreme travel of the valve.

PISTON SPEED OF LOCOMOTIVES.

In determining the proper area for steam and exhaust ports of locomotives it is necessary to ascertain the maximum speed of piston in feet per minute, which may be done according to the following rule:

RULE.—First, multiply the number of feet in one mile (5280) *by the maximum number of miles the locomotive is to run per hour, and divide the product by* 60 (*the number of minutes in one hour*), *and the quotient will give the number of feet the engine travels per minute.*

Second, determine the circumference of the driving wheel by multiplying the constant 3.1416 *by the diameter of the driving wheel in feet, and then divide the speed of the locomotive in feet per minute by the circumference of the driving wheel, and the quotient will give the number of revolutions of driving wheel per minute.*

Third, multiply the number of revolutions of the driving wheel per minute by twice the length of the stroke of the piston in feet, and the product will give the piston speed in feet per minute.

Example.—Let 5280 equal number of feet in one mile.
Let 72 miles equal speed of locomotive per hour.
Let 60 equal number of minutes in one hour.
Let 3.1416 equal a constant.
Let 6 feet equal diameter of driving wheel.
Let 2 feet equal length of stroke of piston.
Let 2 equal number of strokes of piston for each revolution of driving wheel.

Then we have:

$$\frac{(5280\times72)\div60}{3.1416\times6}\times2\times2=1345.72+\text{ feet.}\quad\text{Piston speed per minute.}$$

Performing the operation in the ordinary way, we have:

```
      5280  Number of feet in one mile.
        72  Number of miles locomotive travels per hour.
     -----
     10560
    36960
    ------
60) 380160 (6336  Number of feet locomotive travels per minute.
    360
    ---
     201
     180
     ---
      216
      180
      ---
       360
       360
       ---
```

Next, determining circumference of driving wheel, we have:

```
 3.1416  A constant.
      6 feet.  Diameter of driving wheel.
-------
18.8496 feet.  Circumference of driving wheel.
```

Next, dividing the number of feet the locomotive travels per minute, by the circumference of the driving wheel in feet, we have:

```
18.8496) 6336.0000 (336.13+  Revolutions of driving wheels per minute.
         5654 88
         -------
          681 120
          565 488
          -------
          115 6320
          113 0976
          --------
            2 53440
            1 88496
            -------
              649440
              565488
              ------
```

Finally, multiplying the number of revolutions of driving wheels per minute by twice the length of the piston stroke in feet, we have:

```
         336.13
2×2=          4
        -------
        1344.52 feet.  Piston speed per minute.
```

CHAPTER XXVII.

STEAM BOILER INJECTORS.

The knowledge of the capacity of a moving jet of steam, or other fluid, to produce a vacuum in properly formed ducts, for the purpose of raising air, water or other fluids, and conveying them from one place to another, may be traced back to the time of Venturi, Nicholson and others. Nevertheless, it must be admitted that the eminent French engineer, H. J. GIFFARD, was the first to conceive the idea that the kinetic energy of a moving mass of fluid could be utilized to overcome the static energy of a mass of water under boiler pressure, so that the first mass of fluid would enter such boiler against the resistance of the second.

The terms "kinetic energy" and "static energy" may not be absolutely correct in a mechanical sense in this connection, but they may be used here for the purpose of designating the inherent difference between the conditions of the two masses of fluid under consideration, which will receive notice further on.

THE ESSENTIALLY ACTIVE PARTS OF AN INJECTOR.

First. A steam nozzle through which the operating steam from the boiler enters the injector.

Second. A combining and condensing nozzle, in which the steam and feed water meet, and in which the steam condenses and transmits its dynamic force to the water.

Third. A delivery nozzle in which the maximum velocity of the combined mixture of steam and water is attained.

If these three parts, which are to be found in every injector, are looked upon as the essential components of an injector, the Marquis MANNAURY D'ECTOT must be considered the inventor of the injector. In 1818 he was granted a French patent for a steam jet apparatus, which was capable of raising water from a tank and delivering it into a second tank. After d'Ectot, the French engineers, Pelleton and Bourdon (of monemeter fame), published various inventions of a similar character.

On March 8, 1858, H. J. GIFFARD was granted his first patent for an injector to be used as a boiler feeder. The difference between his apparatus and those of former inventors, consisted mainly of a certain dis-

tance between the outlet orifice from the condensing nozzle and the inlet orifice to the delivery nozzle, which is called the overflow space, and which communicates with the outer atmosphere.

In starting the injector more water, as a rule, enters the apparatus than the injector is capable of delivering against the back pressure of the boiler, and if there were no communications with the atmosphere the injector would "break," that is, refuse to work, and the steam blow back into the tank from which the water is taken; because the steam would naturally follow the line of the least resistance. By providing for an overflow space between the condensing and delivery nozzles, the surplus water is given an opportunity to escape, until the jet of combined steam and water has attained a sufficient amount of velocity and over pressure to open the boiler check valve and deliver the jet into the boiler. After the apparatus has been started and is in operation, the waste of water may be avoided by cutting down the supply until no more water is seen at the overflow.

Giffard's invention, therefore, consisted in the discovery that in order to deliver water through an injector into a boiler, by means of steam taken from the same boiler, the jet must first overflow into the atmosphere. Injectors made prior to Giffard's were, in fact, based upon the same principle as that of the Giffard injector, but were unable to deliver against pressure.

The advantages of this new method of boiler feeding, the simplicity and efficiency of the apparatus, and the comparatively small expense of installation and maintenance, were soon appreciated by steam users, and to-day the injector is among the most popular boiler feeding apparatus in use.

Soon after Giffard's injector had been placed upon the market, the manufacture of injectors became a most important and extensive industry in the United States, as well as in Europe. Hundreds of thousands of injectors have been manufactured for various purposes, and to-day there is hardly a locomotive engine running anywhere which is not provided with at least one injector, though by far the greater majority of them are equipped with two. Nearly every steam vessel in the United States and many stationary plants on land are equipped with injectors as boiler feeders. Hence it is of the utmost importance that every engineer, stationary, marine and locomotive, should thoroughly familiarize himself with the theory and principles which underlie the operation, management and construction of injectors.

Giffard seems to have so thoroughly mastered the laws relating to the action of a moving fluid mass, that the curves and tapers of the nozzles, as constructed by him, have been adopted by all his followers, and are used to-day with scarcely any perceptible change. Whatever

changes or modifications have been made by the numerous manufacturers of boiler-feeding injectors, relate chiefly to constructive details of the bodies of the instruments, and of the different valve mechanisms which control the flow of the steam and of the water. To attempt to describe the numerous types of injectors now in service would be largely a matter of reiteration, and for this reason it will answer the purpose of this chapter to select a few representative examples of modern efficient injectors, and describe their construction and operation, and in this way convey to the minds of engineers and students of steam engineering a clear idea of the construction and operation of boiler feeding injectors generally. We will therefore begin with the Monitor injector, as shown in Figs. 513 and 514. S represents handle of steam spindle; J, handle of lifting jet; W, handle of water valve; H, handle of heater cock; O, overflow; L, line check (Fig. 514). P represents plug for oiler (Fig. 513).

Fig. 515 is a longitudinal sectional view, showing the interior of the injector: 1, body (back part); 2, body (front part); 3, body screw; 4, yoke; 5, yoke gland; 6, yoke packing nut; 7, yoke lock nut; 8, steam-valve disc and nut; 9, steam-valve spindle; 10, steam-valve handle; 11, steam-valve rubber handle; 12, steam-valve top nut; 13, jet-valve disc and nut; 14, jet-valve spindle; 15, jet-valve bonnet and nut; 16, jet-valve gland; 17, jet-valve lever handle; 18, jet-valve top nut; 18*a*, jet tube; 18*b*, lifting nozzle; 19, water valve; 19*a*, eccentric spindle; 20, water-valve bonnet; 23, water-valve lever handle; 25, steam nozzle; 26, intermediate nozzle; 27, condensing nozzle; 28, delivery nozzle; 30, line check; 31, line-check valve; 32, stop ring; 33, overflow nozzle; 33*a*, overflow chamber with nut; 34, heater-cock check; 35, heater-cock bonnet and nut; 36, heater-cock spindle; 37, heater-cock T handle; 38, coupling nut (steam end); 39, coupling nut (water end); 40, coupling nut (delivery end); 38*a*, tail piece (steam end); 39*a*, tail piece (water end); 40*a*, tail piece (delivery end).

This injector is provided with the usual inlet openings for steam and feed water, and an outlet opening for the delivery, which are designated in the illustration (Fig. 515) by arrows, and by the words "steam," "water," "delivery."

The body of the injector consists of two parts, 1 and 2, bolted together, as shown in Figs. 513, 514 and 515. This construction facilitates a thorough inspection and cleaning of the interior of the body whenever such should be deemed necessary or desirable. Part 8, Fig. 515, is the steam valve which controls the flow of steam from the boiler into the nozzles, and is operated by means of the handle, parts 9, 10, 11 and 12. The gland or follower 5 has two arms cast on it,

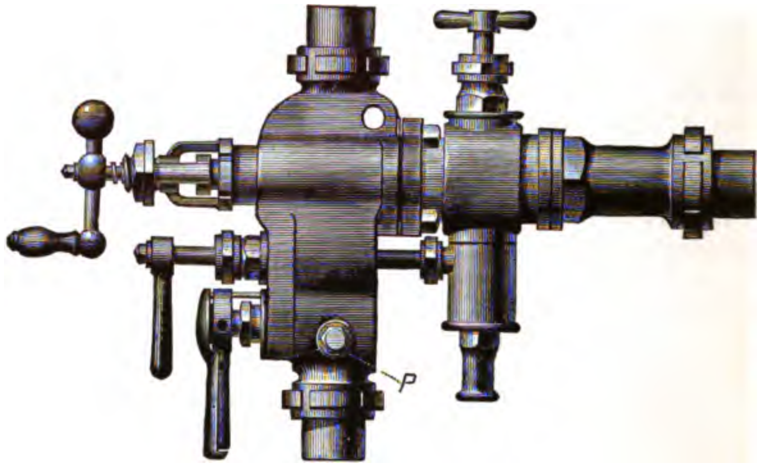


Fig. 513.
EXTERIOR VIEW.

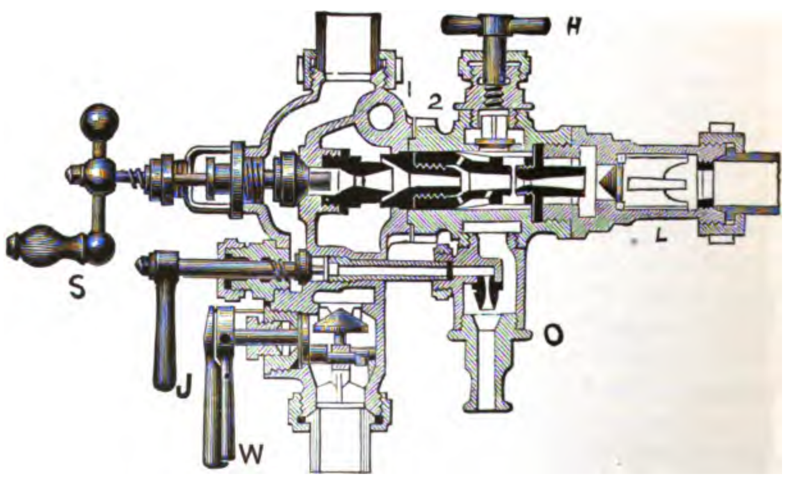


Fig. 514.
INTERIOR VIEW.

passing by the threaded neck of the yoke 4, engaging a circular groove in the packing nut 6. Part 7 represents a lock nut to fix the position of nut 6. By these means the adjusting of the packing is accomplished, uniformly, by a single central nut, in the same manner as in an ordinary stuffing box, without exposing the thread on the spindle to the action of the steam within the chamber.

Parts 18*b* and 33 form an independent lifting mechanism, receiving the operating steam directly from the boiler through tube 18*a*, the inlet to which is controlled by valve 13 and handle 17, and connections 14, 15, 16 and 18.

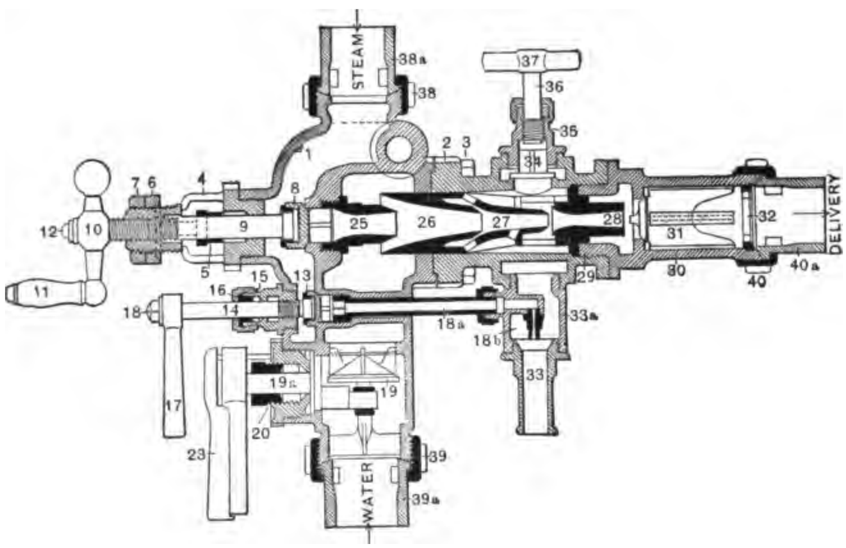


Fig. 515.

Part 19 represents the valve controlling the water inlet by means of the handle 23 and parts 19*a* and 20.

This handle 23 is a double one, and in its upper part is provided with a steel pin, which engages notches milled into the round flange of bonnet 20. In this manner, handle 23 may be placed and retained in any position, and by working the notch, the engineer is enabled to place the handle into any certain position which he may have found most suitable for a certain degree of pressure or amount of delivery.

Part 34 is a check valve, working automatically, and preventing the drawing in of air after the injector has been started.

Part 31 is a check valve, which, in addition to an ordinary main boiler check valve, placed between the injector and the boiler, prevents the back flow of water from the boiler, while the injector is not in operation.

Parts 25, 26, 27 and 28 are the steam, intermediate, condensing, and delivery nozzles, respectively.

The operation of the apparatus may be described as follows:

Connection having been made with the boiler at the points marked "steam" and "delivery," and with the water supply at the point marked "water," jet valve, part 13, is opened first. This will cause steam to rush through the jet tube 18*a*, lifting nozzle 18*b*, and overflow nozzle 33 into the atmosphere, and will produce a vacuum in the overflow chamber 33*a*. It will follow, as a natural result, that the atmospheric pressure will force the water up the water tube, through intermediate and condensing nozzles 26 and 27, lifting the heater-cock check 34, and throwing water through overflow nozzle 33. Water appearing at this overflow, steam valve 8 is opened, and then jet valve 13 is closed. The boiler steam passing through steam nozzle 25 will mix with the water in nozzles 26 and 27, gradually accelerating the velocity of the mixture, until it becomes sufficiently great to pass through delivery nozzle 28, and to open both check valve 31 and the boiler check valve.

The quantity of water to be allowed may be regulated by means of valve 19. At high degrees of steam pressure, no regulation will be necessary, because the injector will then readily take up and deliver all the water supplied to it. At low pressures, however, more water will reach the injector than it is capable of delivering, and in such cases the supply must be regulated by the water valve 19.

In locomotive practice it becomes occasionally necessary to warm the feed water in the tank. This can be done with the aid of the injector, by screwing down heater-cock spindle 36 and 37. The opening of heater-cock check 34 will thus be prevented, and if steam valve 8 is kept open steam will flow back through the water passages into the tank.

AUTOMATIC INJECTORS.

Fig. 516 is an exterior view, and Fig. 517 is an interior view of an automatic injector. It is automatic in that no regulation of the water supply is required. It will lift the water and force it into the boiler with a single steam jet. This class of injectors is largely used as feed-water feeders for stationary boilers, and for which they are peculiarly adapted. The Monitor, the injector previously described, however, has the advantage of being able to force water against a greater pressure than the type of injector shown in Figs. 516 and 517. The principle upon which the latter works does not differ materially from other injectors. Therefore, when an engineer has familiarized himself with the construction and manner of operating any of the standard injectors he will experience little or no difficulty in operating any injector.

Fig. 516.

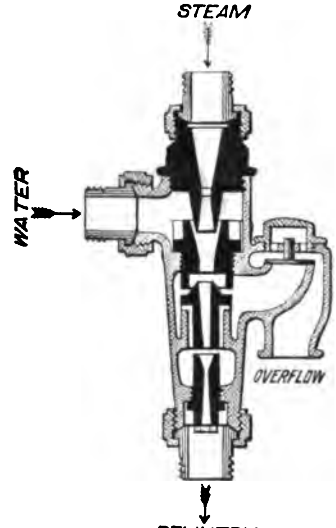


Fig 517.

DOUBLE JET INJECTORS.

Fig. 518 is an exterior view, and Fig. 519 an interior view of the World injector, showing all of its working parts. This is a double jet instrument and works somewhat similar to the Monitor, giving about the same results as to over pressure and long and short lift.

When the valves C and D are closed the valve C comes in contact with the valve seat at the bottom of the counterbore in steam nozzle K, so that before the valve can present a steam opening into the steam nozzle K it will have to be withdrawn from the counterbore. This construction is for the purpose of allowing the valve D to be withdrawn from its seat in the steam nozzle J, and lifting water to space 1 and forcing it through the nozzle F into spaces 2 and 3, and overflowing through the overflow nozzle 8, before the steam valve C is opened.

When both steam valves are closed the handle L is up, as shown in Fig. 518; when in operation the handle is down, as shown in Fig. 519.

To start the injector the throttle valve is opened and the steam let into the passageway G H H and I. The handle L is turned toward the left about a quarter of a turn, and the valve D is withdrawn from its seat in steam nozzle J, which forms a vacuum in the passageway 1, and the atmospheric pressure forces the water up and fills the space 1, from there it is syphoned in and through the nozzle F into spaces 2 and 3. The valve B is open and the water overflows into 7 and 8 from

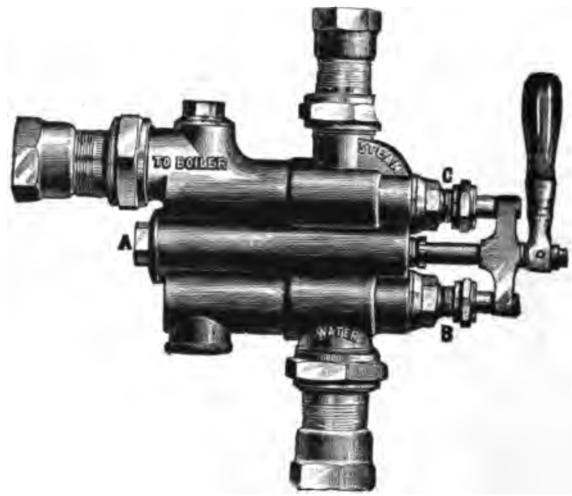


Fig. 518.

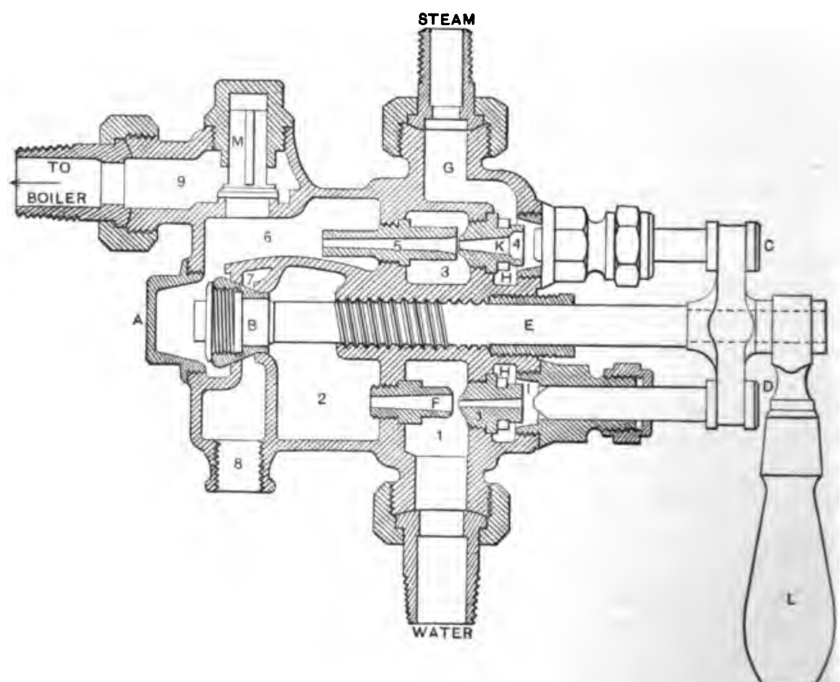


Fig. 519.

2 direct, and from 3 through the nozzle 5 into space 6, and thence into 7 and 8 and out into the atmosphere. The valve B fits loosely on the end of the central stem E, similar to a globe valve. The small end of the valve is plug shaped, and made to fit the opening from 2 into 7 and 8, so that the overflow from 2 may be closed, while that from 6 into 7 and 8 is still open. The valve B is bored out at its small end to fit the central stem at B, and counterbored to fit a collar on the end of the

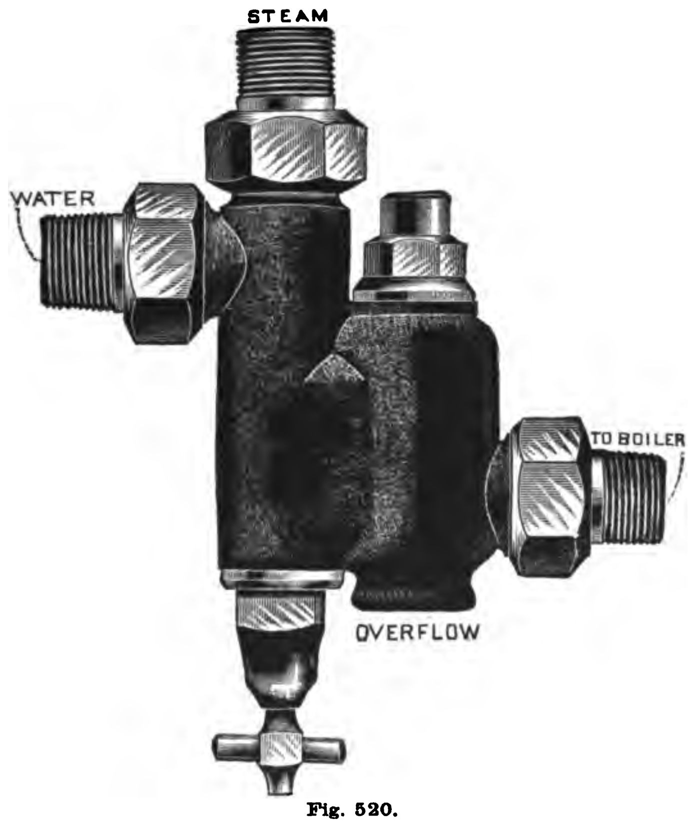


Fig. 520.

central stem. The valve is slipped onto the central stem from the handle end before the stem is put into position. A plug is screwed into the valve close to the collar on the end of the central stem, so as to allow the stem to work loosely in the valve, and yet not have play enough to allow the valve to tip downward or get out of line.

When water appears at the overflow the handle L may be turned far enough to admit steam into the steam nozzle K, when the water

will be drawn from 3 and forced through the nozzle 5 into 6, and from there into 7 and 8. Then by moving the handle L as far as it will go, the overflow valve B will be closed, as shown in Fig. 519, and the water will flow by the check valve M into discharge pipe 9, and thence into the boiler.

In the construction of this injector the utmost simplicity has been strictly observed; at the same time it presents a most ingenious arrangement of all the parts so as to produce the highest standard of efficiency in steam boiler feeding.

The injector, shown in Figs. 520 and 521, is a single jet device for feeding boilers. It is automatic in its operation; that is, it will start itself when the steam is turned on without any adjustment of valves or other parts. If, from any cause the water should be cut off, the injector will start again the moment water returns to it.

With this kind of device not quite so much over pressure can be attained as with the double jet injector already described. With the double jet injector the discharge water can be forced, without difficulty, against a pressure of 225 pounds per square inch, with a steam pressure of 100 pounds per square inch; while with the single jet injectors water can be forced against a pressure of 140 pounds per square inch, with a steam pressure of 80 pounds per square inch. This injector will work on a lift of 3 feet, with from 15 to 145 pounds of steam; with a longer lift it will not do quite as well, although with 80 pounds of steam pressure it will lift water about 20 feet.

When the throttle valve is opened the steam passes through the steam nozzle 1 (Fig. 521) into nozzle 2, syphoning the water from inlet A and forcing it through 2 into 3, at the top of which it overflows into space B until the injector is in full operation, when all of the water flowing to it is taken up and discharged through 2 3 4 into 5, thence into 6 to 7 and thence into the boiler. The water overflowing into B passes down to C, thence under the valve 8 into D, and down around 6 into E, and thence into the atmosphere; which overflow, however, will cease as soon as the injector is in full operation, as it will then take up all the water flowing through the intermediate water nozzle 2.

This automatic injector, like that shown in Figs. 516 and 517, is used largely in feeding stationary boilers, and boilers where excessive high steam pressure is not carried.

CAUSES WHICH PREVENT INJECTORS WORKING.

One of the main causes which prevents an injector working is air in the suction pipe, caused by leakages in the pipe or improper packing, or lack of proper packing in stem of valve in suction pipe. When injectors are used on steamers, faulty construction of supply tank fre-

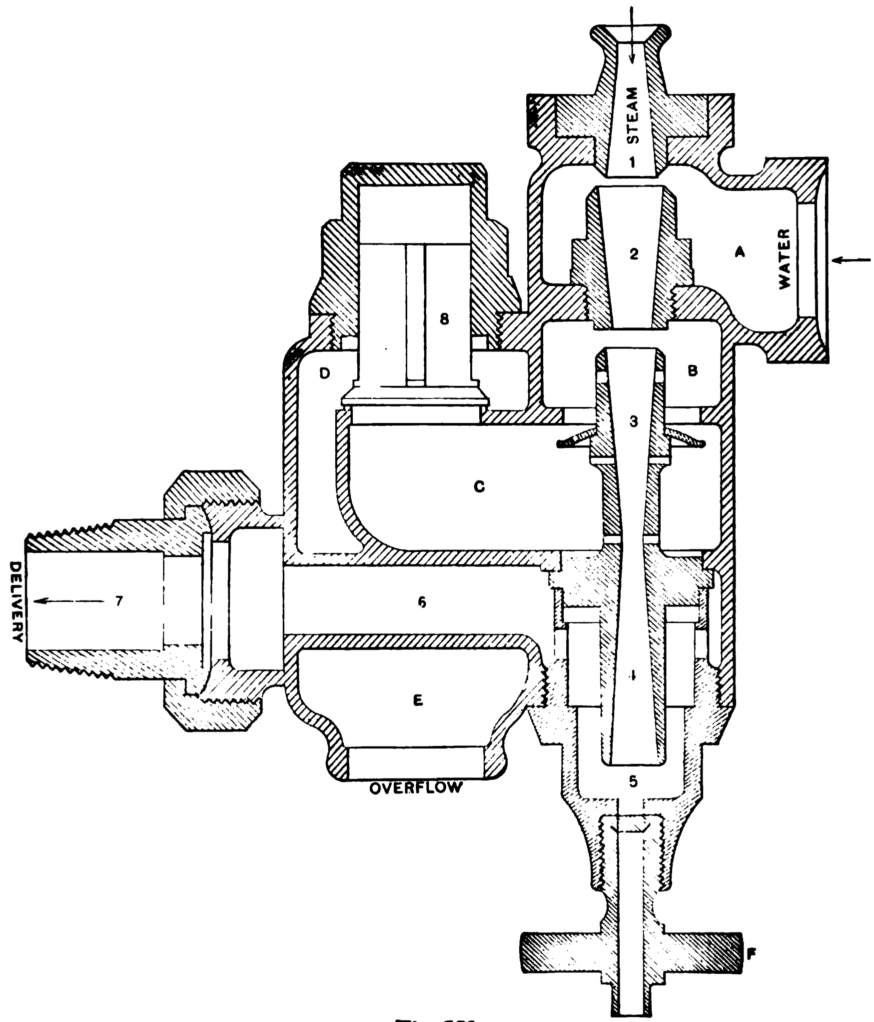


Fig. 521.

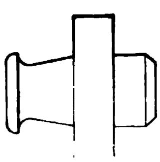


Fig. 522.
STEAM NOZZLE.

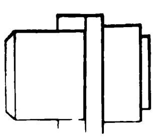


Fig. 523.
INTERMEDIATE NOZZLE.

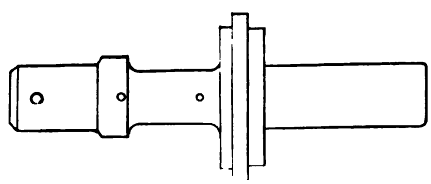


Fig. 524.
CONDENSING NOZZLE.

quently interrupts the working of the injector. The lower end of the injector suction pipe should always be surrounded and submerged in a solid body of water and entirely free from air.

Fig. 525 is a perspective view of a properly constructed supply tank for injectors. A A′ represent the supply pipes leading into the tank T, and D D′ represent short pieces of pipe extending upward from and attached to the inner ends of the supply pipes; allowing the supply water to overflow from the upper ends of the short pipes. In case any air should pass into the tank with the water it will escape through the air pipe C, in the top and at the opposite end of the tank. B represents the suction pipe leading to the injector; the lower end extends

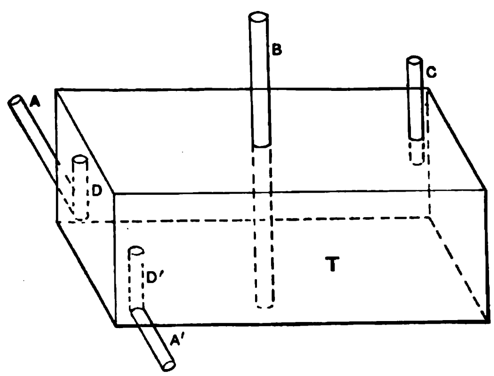


Fig 525.

SUPPLY TANK FOR STEAMBOATS.

down into the water, to within a short distance from the bottom of the tank. This tank and connections present a very simple arrangement, and when constructed, as shown in Fig. 525, removes entirely the main cause which prevents injectors working properly.

Injectors work best with dry steam; for that reason the steam supply pipe should be attached to the highest point in the steam drum or dome.

Sediment and scale in nozzles also are frequent causes of injectors failing to work properly. The nozzles should be removed occasionally and thoroughly cleaned.

When the injector fails to get water, the supply may be cut off by absence of water at the source; strainer clogged up; the supply pipe, hose or valve stopped up; supply pipe or water too hot; lack of sufficient pressure for the amount of lift; leak in the supply pipe, its connections or its valve—the latter three are the most frequent causes.

When the injector gets water and fails to force it into the boiler, it may be that there is too much or too little water, dirt in the

delivery tube, faulty check valve, obstruction between injectors and boiler check valve.

When the injector starts but "breaks," the supply of water may not be properly regulated; the supply pipe may be admitting air to the injector; there may be some obstruction to the free delivery of the water on account of dirt in the delivery tube, faulty check valve, or obstruction between injector and boiler check valve. Sometimes a globe valve used on the supply connection has a loose disc, and after starting the disc is drawn down, partially closing the valve, which is equivalent to giving the injector too little water. This only occurs, however, when starting with low steam. To remedy this take the valve off and reverse it.

Having described the practical operation of injectors and the manner of their construction, as well as the causes which tend to obstruct their working, the next subject for consideration will be the theory of the injector.

SPECULATIVE THEORY OF THE INJECTOR.

WHY AN INJECTOR WORKS.

The reason why and how an injector works may be mathematically explained as follows:

For the purpose of making deductions as simple as possible, it will be assumed with Combes, that steam flows into the atmosphere with the same density and under the same pressure which it had within the boiler. Under such a condition the velocity of steam issuing from a certain orifice, and not taking into account the influence of the form or shape of the orifice, may be computed from the following formula:

$$V = \sqrt{2\,g\,\frac{P-p}{q}}$$

In which V equals the velocity of steam in feet per second.
- 2 equals a constant.
- g equals a co-efficient of acceleration (32.16 feet).
- P equals steam pressure per square foot in the boiler.
- p equals atmospheric pressure per square foot.
- q equals specific gravity of steam (weight of one cubic foot).

To dwell upon a concrete example, we will assume the boiler pressure to be 70 pounds per square inch, and that P in the equation

represents the boiler pressure per square foot; p the atmospheric pressure per square foot, and q the weight of a cubic foot of steam at 70 pounds pressure per square inch equals 0.168 pounds, therefore,

$$\frac{P-p}{q}=\frac{(144\times 70)-(144\times 14.7)}{0.168}=47400$$

And $\qquad V=\sqrt{2\times 32.16\times 47400}=1746+$ feet per second.

As the student may not be familiar with algebra, the formula

$$V=\sqrt{2\,g\cdot\frac{P-p}{q}}$$

will be explained and reduced to simple arithmetic.

The operation to be performed, as indicated by the formula, is that the value of p is to be subtracted from the value of P, and the remainder divided by the value of q, and the quotient to be employed to multiply the product of 2 multiplied by the value of g, and finally that the square root is to be extracted of the last product, and the answer will equal V; which represents the velocity of steam in feet per second.

The value of g is 32.16, and that represents the velocity a falling body attains in feet at the end of the first second, when falling in a vacuum. The distance a body will fall in one second in vacuum is 16.083 feet; but when it has reached that distance its velocity will be at the rate of 32.16 feet per second; in other words, if it would continue at the rate of speed it had attained at the end of its first second of fall, without any variation in speed, it would travel a distance of 32.16 feet per second. The velocity of 32.16 feet is therefore taken as a unit or measure of force of gravity, and is denoted by the letter g, and this in the formula is multiplied by 2, which equals 64.32.

P in the formula represents the steam pressure per square foot, no matter what that pressure may be. In the case under consideration it is assumed to be 70 pounds per square inch, so that in the equation the steam pressure per square foot would be $144\times 70=10080$ pounds. In the formula p represents the atmospheric pressure per square foot. In the case under consideration it is assumed to be 14.7 pounds per square inch, so that in the equation the atmospheric pressure per square foot would be $144\times 14.7=2116.8$ pounds. q, like the steam pressure, is a variable quantity. It represents the weight, in pounds, of a cubic foot of steam at any given pressure. As the weight of a cubic foot of steam varies with the pressure, recourse must therefore be had to the steam tables to ascertain the weight of a cubic foot of steam for any given pressure per square inch. In the present case the

steam pressure is given at 70 pounds per square inch, and by reference to the table of "Properties of Saturated Steam," it will be found that the weight of a cubic foot of steam for that pressure is 0.168 pounds, and this weight, whatever it may be, is represented by the letter q. We are now prepared to demonstrate the formula arithmetically.

VELOCITY OF STEAM FLOWING INTO THE ATMOSPHERE

RULE.—First, multiply the number of square inches in a square foot (144) *by the given steam pressure per square inch, in pounds, and call the product "Product No. 1."*

Second, multiply the number of square inches in a square foot (144) *by the atmospheric pressure per square inch* (14.7 *pounds*), *and call the product "Product No. 2."*

Third, subtract "Product No. 2" from "Product No. 1," and call the answer "The Remainder."

Fourth, ascertain the weight of a cubic foot of steam due to the given steam pressure, as found in the table of "Properties of Saturated Steam," and divide "The Remainder" by the weight of a cubic foot of steam, and call the answer "The Quotient."

Fifth, multiply the velocity per second a body falling in vacuum has attained at the end of one second of time (32.16 *feet*) *by the constant* 2, *and call the product "Product No. 3."*

Sixth, multiply "The Quotient" by "Product No. 3," and then extract the square root of the last product, and the root thus found will give the velocity of steam per second flowing into the atmosphere.

Example.—Let 144 inches equal a square foot.

Let 70 pounds equal given steam pressure per square inch.

Let 14.7 pounds equal atmospheric pressure per square inch.

Let .168 pounds equal weight of a cubic foot of steam at 70 lbs. pressure.

Let 32.16 feet equal velocity per second a body falling in vacuum has attained at the end of the first second.

Let 2 equal a constant.

Then we have:

$$\sqrt{\left(\frac{144\times70-144\times14.7}{.168}\right)\times32.16\times2}=1746+ \text{ feet. Velocity of steam per second.}$$

Performing the operation in the ordinary way, we have:

```
  144   Number of square inches in a square foot.
   70   Pounds of steam pressure per square inch.
-----
10080   "Product No. 1."

  144   Number of square inches in a square foot.
 14.7   Pounds of atmospheric pressure per square inch.
 ----
 1008
 576
144
------
2116.8  "Product No. 2."
```

Next, subtracting "Product No. 2" from "Product No. 1," we have:

```
10080.0  "Product No. 1."
 2116.8  "Product No. 2."
-------
 7963.2  "The Remainder."
```

Next, dividing "The Remainder" by the weight of a cubic foot of steam due to 70 lbs. steam pressure per square inch, we have:

```
.168) 7963.200 (47400  "The Quotient"
      672
      ----
      1243
      1176
      -----
        672
        672
        ---
         00
```

Next, multiplying the velocity a body falling in vacuum has attained at the end of the first second, by the constant 2, we have:

```
32.16
    2
-----
64.32  "Product No. 3."
```

Next, multiplying "The Quotient" by "Product No. 3," we have:

```
     47400  "The Quotient."
     64.32  "Product No. 3."
    ------
    948 00
  14220 0
 189600
284400
----------
3048768.00  The last product.
```

Finally, extracting the square root of the last product, we have:

$$
\begin{array}{rl}
 & \dot{3}0\dot{4}8\dot{7}6\dot{8}\ (1746+ \text{ feet.} \\
 & 1 \\
27) & 204 \\
 & 189 \\
344) & 1587 \\
 & 1376 \\
3486) & 21168 \\
 & 20916
\end{array}
$$

Velocity of steam flowing into the atmosphere per second.

PROPORTION OF WATER TO THAT OF STEAM.

Before this steam can issue from the orifice it comes in contact with water, by which it is condensed, and with which it forms a mixed jet of steam and water. The velocity of this water which mixes with the steam is very slow, as compared with that of the steam itself, and may therefore be entirely ignored, without materially affecting the results in view, so that we arrive at this equation:

$$(m+M)v=mV$$

in which M equals mass of water mixing with the steam.
m equals mass of steam issuing per time unit.
V equals velocity of steam figured above.
v equals velocity of the mixed jet.

$$v=V\frac{m}{m+M} \quad (1)$$

The first thing is to determine the value of the different letters in the equation. This is done by ascertaining the temperature of the water flowing to the injector, the total heat units of the steam due its boiler pressure per square inch, and the temperature of the mixture; that is the temperature of the water discharged into the boiler by the injector. In other words, the temperature of the water after the steam is mixed with it.

As the boiler pressure in the case under consideration has been taken at 70 lbs. per square inch, by reference to the steam table it will be found that the total heat units in one pound of steam at that pressure is 1206, and as the temperature, we will say, of the water after being mixed with the steam has been raised from 75 degrees to 160 degrees, we deduct the degrees of temperature to which the water has been raised from the heat units in the steam due the pressure, which in this case is 70 lbs. per square inch, and the heat units due that

pressure will be found by reference to the steam table to be 1206, and deducting the degrees of temperature of the mixture (160), we have:

$$\begin{array}{r} 1206 \\ 160 \\ \hline 1046 \end{array} \text{ "Remainder No. 1."}$$

In the next place we deduct the degrees of temperature of the water flowing to the injector from the degrees of temperature of the mixture, or water flowing from the injector, which in this case is 75 degrees for the former and 160 degrees for the latter, and we have:

$$\begin{array}{r} 160 \\ 75 \\ \hline 85 \end{array} \text{ "Remainder No. 2."}$$

We next divide "Remainder No. 1" by "Remainder No. 2" and the quotient will give the amount or mass of water discharged by the injector as compared with that of the steam, and we have:

$$\begin{array}{r} 85)\ 1046\ (12+ \\ 85 \\ \hline 196 \\ 170 \\ \hline \end{array} \text{ This represents the proportion of the water to 1 of steam.}$$

And hence 1 represents the value of m and 12 represents the value of M in equation No. 1.

We are now prepared to demonstrate equation No. 1 arithmetically.

TO DETERMINE THE VALUE OF m AND M IN EQUATION No. 1.

RULE.—First, subtract the number of degrees of temperature of the water discharged by the injector from the total heat units in one pound of steam due the given pressure, and call the remainder "Remainder No. 1."

Second, subtract the number of degrees of the water flowing to the injector from the number of degrees of temperature of the water discharged by the injector, and call the remainder "Remainder No. 2."

Third, divide "Remainder No. 1" *by "Remainder No.* 2," *and the quotient will give the proportion of the water to* 1 *of steam contained in the mixture, and consequently the value of m and* M *in equation No.* 1.

Example.—Let 1206 units of heat equal the number in one pound of steam at 70 pounds pressure per square inch.

Let 160 degrees equal temperature of mass of water discharged by the injector.

Let 75 degrees equal temperature of water flowing to the injector.

Then we have: $$\frac{1206-160}{160-75}=12+ \text{ Proportion of water to 1 of steam.}$$

Performing the operation in the ordinary way, we have.

$$\begin{array}{r} 1206 \\ 160 \\ \hline 1046 \end{array} \text{ "Remainder No. 1."}$$

Next we have:

$$\begin{array}{r} 160 \\ 75 \\ \hline 85 \end{array} \text{ "Remainder No. 2."}$$

Finally, dividing "Remainder No. 1" by "Remainder No. 2," we have

$$\begin{array}{l} 85\,)1046\,(\,12+ \text{ Proportion of water to 1 of steam.} \\ \quad\;\; 85 \\ \quad\;\; \overline{196} \\ \quad\;\; 170 \\ \quad\;\; \overline{} \end{array}$$

And $$\frac{12}{12}=1$$

and hence, 12 represents the value of M and 1 represents the value of m in equation No. 1.

TO DETERMINE THE VELOCITY OF THE MIXTURE DISCHARGED BY THE INJECTOR.

Before laying down the rule for determining the velocity of the mixture, it will be well to remind the student that this velocity does not represent the velocity with which the mixture flows into the boiler, but it represents the velocity with which it would flow into the atmosphere.

RULE.—Add the proportion of steam to the proportion of water contained in the mixture, and divide the proportion of steam by the sum, and multiply the quotient by the velocity of the steam per second; the product will give the velocity of the mixture in feet per second.

Example.—Let 1 represent the proportion of steam to water contained in the mixture.

Let 12 equal the proportion of water contained in the mixture.

Let 1746 feet equal velocity of steam flowing into the atmosphere per second at 70 pounds pressure per square inch.

Then we have:

$$\frac{1}{1+12}\times 1746 = 134.3+ \text{ feet per second.}$$ **Velocity with which the moving jet from the injector will flow into the atmosphere.**

Performing the operation in the ordinary way, we have:

12	**Proportion of water to steam.**
1	**Proportion of steam to water.**
13	**The sum.**

Next, dividing the proportion of steam by the sum of the steam and the water, we have:

```
13) 1.00 (0.07692+   The quotient.
     91
     --
      90
      78
      ---
      120
      117
      ---
        30
        26
        --
```

Finally, multiplying the quotient by the velocity of steam per second flowing into the atmosphere under a pressure of 70 pounds per square inch, we have:

```
      .07692
        1746
      ------
       46152
      3 0768
     53 844
     76 92
   ---------
   134.30232 feet per second.
```

Velocity with which the moving jet from the injector will flow into the atmosphere.

If this velocity is greater than that with which the water would flow from the boiler, under the same pressure of 70 pounds per square inch, it becomes evident that the moving fluid, connected to the boiler by suitable piping, will enter it.

The velocity with which the water would issue from the boiler may be calculated by the following formula:

$V' = \sqrt{2\,g\,h}$, in which g is again 32.16, and h equals the height of a water column corresponding to the pressure of 70 pounds per square inch (161 feet, in round numbers) from which it follows that $V' = 101.76+$ feet per second. Again we will reduce the formula to simple arithmetic and give the rule for performing the operation.

TO DETERMINE THE VELOCITY OF WATER FLOWING FROM THE BOILER UNDER A GIVEN PRESSURE.

First determine the height of a column of water in feet corresponding to the given pressure per square inch.

RULE.—Divide the number of cubic inches in a cubic foot of water (1728) *by* 12, *and divide the quotient by the weight of a cubic foot of water* (62.5 *pounds*); *then multiply the last quotient by the given steam pressure* (70 *pounds in this case*), *and the product will give the height of a column of water in feet corresponding to the given pressure per square inch.*

Example.—Let 1728 equal number of cubic inches in a cubic foot of water.
Let 12 equal a constant.
Let 62.5 pounds equal weight of a cubic foot of water.
Let 70 pounds equal given steam pressure per square inch.

Then we have:

$$\left(\frac{1728}{12} \div 62.5\right) \times 70 = 161.280 \text{ feet.}$$

Height of a column of water corresponding to a pressure of 70 pounds per square inch.

Performing the operation in the ordinary way, we have:

```
12) 1728 (144   The quotient.
    12
    --
     52
     48
     --
      48
      48
      --
```

Next, dividing the quotient by the weight of a cubic foot of water, we have:

```
62.5) 144.0 (2.304   The last quotient.
      125 0
      -----
       19 00
       18 75
       -----
         2500
         2500
         ----
```

Finally, multiplying the last quotient by the given steam pressure per square inch, we have:

```
    2.304
       70
  -------
  161.280 feet.
```

Height of a column of water corresponding to a pressure of 70 pounds per square inch.

Then, to determine the velocity of water flowing from a boiler under a given pressure, we proceed as follows:

RULE.—Multiply the velocity in feet that a body falling in vacuum will attain at the end of the first second of time (32.16 *feet per second*) *by* 2, *and multiply the product by the height in feet of a column of water corresponding to the given steam pressure per square inch* (70 *pounds in this case*), *and then extract the square root of the product, and the answer will give the velocity, in feet per second, the water will flow from the boiler.*

Example.—Let 32.16 feet equal velocity per second a body falling in vacuum will attain at the end of the first second.
Let 2 equal a constant.
Let 161 feet equal height of a column of water corresponding to a pressure of 70 lbs. per square inch.

Then we have:

$$\sqrt{32.16 \times 2 \times 161} = 101.76+ \text{ feet. Velocity per second.}$$

Performing the operation in the ordinary way, we have:

$$\begin{array}{rl} 32.16 & \text{Velocity of falling body in feet.} \\ 2 & \text{A constant.} \\ \hline 64.32 & \\ 161 & \text{feet. Height of a column of water corresponding to a pressure of 70 lbs. per square inch.} \\ \hline 64\,32 & \\ 3859\,2 & \\ 6432 & \\ \hline 10355.52 & \text{The product.} \end{array}$$

Finally, extracting the square root of the product, we have:

$$\begin{array}{rl} & \dot{1}0\dot{3}5\dot{5}.\dot{5}2\ (101.76+ \text{ feet. Velocity per second with which water will issue from a boiler with a pressure of 70 lbs. per square inch.} \\ & 1 \\ \hline 201) & 0355 \\ & 201 \\ \hline 2027) & 154\,52 \\ & 141\,89 \\ \hline 20346) & 12\,6300 \\ & 12\,2076 \\ \hline \end{array}$$

Therefore, as the water flowing from the injector, under a steam pressure of 70 lbs. per square inch, has a velocity of 134.3 feet per second, and the water flowing from the boiler under a pressure of 70 lbs.

per square inch has a velocity of 101.76 feet per second, the difference between the two velocities clearly explains the action of the injector; while it also shows that, theoretically, the quantity of water delivered by an injector may be twelve times the quantity of steam consumed for the purpose. The actual quantity delivered will, however, be materially less. The friction of the steam and of the mixture of steam and water, within the nozzles, and within the pipe conduits, will reduce the velocity of the jet, and this, taken in connection with imperfect condensation, and with more or less air finding its way into the instrument, will tend toward reducing its delivering capacity.

It would probably lead too far for the purpose of an elementary work of this kind, to enter minutely into the details of the very intricate item of injector theory, as treated by various authorities; especially in view of the fact that there is no general or universally applicable theory or formula by which the proportions of injector nozzles, as to the areas of orifices, tapers and relative positions could be computed. The conditions upon which the proper function of an injector depends are so manifold that it is impossible to combine them in one comprehensive formula.

DELIVERING CAPACITY OF INJECTORS.

The German scientist, Dr. Grashof, who wrote extensively on this subject, and who is regarded as high authority, gives the following formula for the delivering capacity, C, of an injector, in litres or kilograms:

$$C = A\, d^2 \sqrt{p-1}$$

in which d equals the smallest diameter of the delivery nozzle, p equals steam pressure in atmospheres, A is a co-efficient dependent upon various conditions, but independent of the size of the apparatus. The quotient $\frac{C}{d^2}$ is a measure for the capacity of the injector per square millimeter of the smallest delivery opening, and may therefore be used for the purpose of ascertaining the comparative capacities of different makes of injectors.

Dr. Grashof also gives the size of the steam nozzle, and the area of the water inlet around the steam nozzle, in the following formula:

$$A = \frac{G'}{y'\, v'} = \frac{G^2}{x\, y' v'}$$

$$A' = \frac{G^2}{y\, v^2}$$

in which A equals area of steam nozzle.

A' equals area of water inlet opening around the steam nozzle.
G' equals weight of steam issuing per second.
v' equals its velocity.
G^2 equals weight of water entering per second.
y' equals specific gravity of the mixture of steam and water.
x equals the proportion $\frac{G^2}{G'}$
y equals the specific gravity of the water.
v^2 equals the velocity with which the water passes by the steam nozzle.

It is evident that these formulæ are not directly applicable, as they depend chiefly upon the velocity and density of the steam, which naturally varies according to pressure. But even if they were directly applicable, they give no information as to taper, length and relative position of the nozzles, which items are fully as important as the areas themselves, and they are, and are likely to remain, largely matters of experimental research.

The efficiency of the injector as a boiler feeder has been fully demonstrated without the aid of any of the advanced theories. Engineers have long been skeptical regarding the reliability of injectors and their ability to force water successfully against a pressure per square inch equal to that of the pressure of the steam supplied to the injector.

Experiments made, by the author of this work, with an injector constructed upon the principle shown in Figs. 513, 514, 518 and 519 with a steam pressure of 80 pounds per square inch in the boiler which supplied the injector, the hydrostatic pressure in a battery of three boilers was forced up to 180 pounds to the square inch. And with a steam pressure of 115 pounds per square inch the hydrostatic pressure in the battery of three boilers was forced up to 225 pounds per square inch. The final test was made with 150 pounds steam pressure, which was much greater than was necessary, and the hydrostatic pressure in the battery of three boilers was forced up to 265 pounds to the square inch without any break in the jet, thus proving conclusively that the injector may be implicitly relied upon as a thoroughly reliable and efficient boiler feeder.

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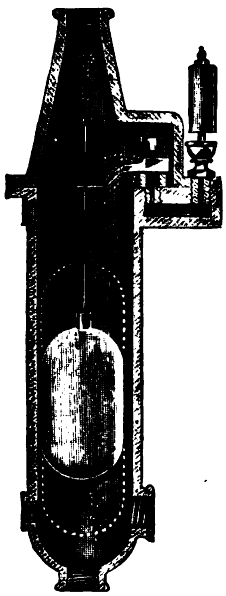

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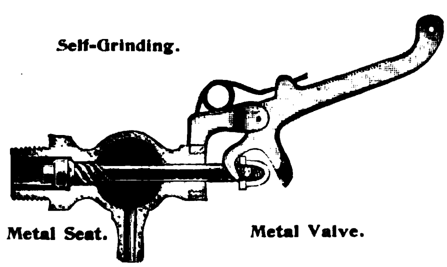

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SEE PAGES Nos. 483, 498, 512, 518 AND 515.